AUG 7 1939

Heating · Piping and Air Conditioning

• Installation
on•Maintenance
DUSTRY AND
E BUILDINGS

GUST, 1939

ents—Page 27

—Journal Section of
n Society of Heating
ntilating Engineers

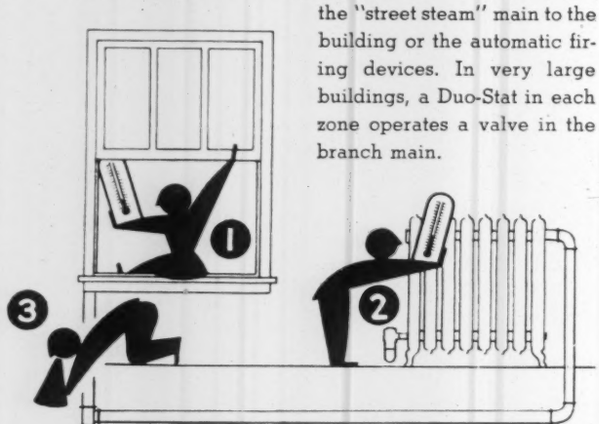
HAPPY OWNERS + SATISFIED TENANTS

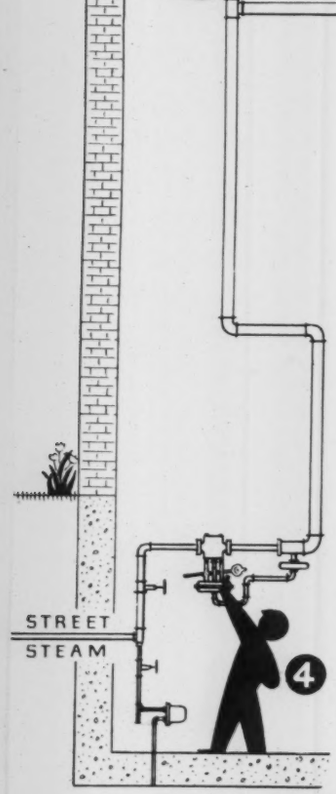
ECONOMY

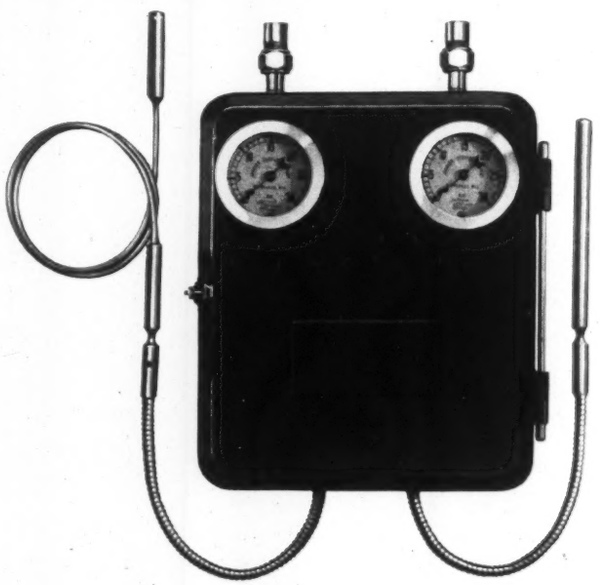
COMFORT

HERE'S HOW IT'S DONE . . .

The *Johnson* Duo-Stat performs all of the operations indicated below: (1) Senses outdoor temperature; (2) Notes the temperature of the heating system; (3) "Telegraphs" to the source of heat the proper relationship between those two temperatures; (4) Directly controls the "street steam" main to the building or the automatic firing devices. In very large buildings, a Duo-Stat in each zone operates a valve in the branch main.







JOHNSON DUO-STATS

"CHANGE THE SIZE" OF HEATING SYSTEMS TO FIT THE OUTDOOR TEMPERATURE

Simple—The principle of the *Johnson* Duo-Stat is fundamentally sound. Radiators are filled with steam only when lowest outdoor temperatures are encountered; partially filled, in proper proportions, when only a part of their total heating capacity is required. (In hot water heating, the temperature of the water is varied in accordance with outdoor temperature.) The Duo-Stat, at the "last radiator" in the building or heating zone, senses the temperature of the heating system and the outdoor temperature, balances one against the other to secure just the right amount of heat. More than ten years of successful Duo-Stat experience prove the logic of this simple principle.

Economical—Room temperatures, with *Johnson* Duo-Stat control, are surprisingly uniform. Tenant satisfaction is assured. And owners are happy, too, because fuel savings are inevita more than 25%, in many cases. The list o hundreds of *Johnson* Duo-Stat installations s second and third, and even more, orders owners who operate more than one building more convincing proof of satisfaction and econ could be offered as a recommendation!

Investigate Now—*Johnson* Duo-Stats ma installed in existing or in new buildings equal facility. Investigate this *tried and p* weather compensating heat control, the de that actually measures the temperature o heating system and maintains the proper ba with conditions outdoors. A *Johnson* sales e neer will make a survey and recommenda without obligation. Act now, before the next ing season. **Ask for information** TODAY!

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JOHNSON SERVICE COMPA

MILWAUKEE, WIS., AND DIRECT BRANCH OFFICES IN PRINCIPAL CI

The Editor's Page

■ **Dollar volume** of residential building contracts let during the first six months of this year was greater than for any similar period since 1929, according to F. W. Dodge Corp. In the 37 states east of the Rocky Mountains, the six months' contract total was $644,527,000, which was 61 per cent greater than the total for the first half of 1938, and 25 per cent greater than the total for the first half of 1937. Out of a $245,000,000 increase over the 1938 period, $181,000,000 represented the increase in private residential building and $64,000,000 measured the increase in public housing projects. All sections of the country participated in the residential building increases, with increments over the first half of 1938 ranging from 19 per cent in New England to 125 per cent in the Cleveland area. . . . Non-residential building also increased over the 1938 period, with a total of $516,579,000 compared with $433,220,000 last year. This was a 19 per cent increase. In the non-residential classification, public contract volume increased 42 per cent and private contracts increased 2 per cent. Heavy engineering contracts (public works and utilities projects) had a contract total of $538,258,000, nearly 17 per cent greater than the corresponding 1938 figure. The combined result of all these building and engineering gains was a six months' construction total of $1,699,364,000, a 31 per cent increase over the first half of 1938. . . . Commenting on the construction record and outlook, Thomas S. Holden, vice-president in charge of the statistical division of F. W. Dodge Corp., said: "Present indications point to a good construction volume during the remainder of the year, although comparisons with the corresponding months of 1938 will be less spectacular than in the first six months. The second half of 1938 was a period of steadily mounting volume after a slack period in the first half, whereas the month-by-month trend this year is following the more usual seasoned pattern. Residential building, both private and public, gives every indication of continuing at a satisfactory rate; non-residential building is likely to taper off somewhat due to lessened public expenditures. Heavy construction, largely supported by federal funds, is also likely to continue tapering off. Even with an ever-narrowing spread between 1939 and 1938 monthly totals from now on, the year will probably close as the sixth consecutive one of construction volume increases."

■ **The editors** wish to acknowledge with thanks a note from Percy Wilbourn of Carey & Wilbourn, ventilation contractors, who writes "A continuous subscriber since before your first publication wishes you a continued and greater success."

■ **Church attendance** in Estill, S. C., has taken a sudden jump, Carrier Corp. reports, since local pastors have been holding joint church services in a new air conditioned theater. . . . Estill is reputed to be the smallest town in the U. S. boasting an air conditioned movie house. . . . Air conditioning may some day be used in the farmer's field, for experimental work of the Minneapolis-Moline Power Implement Co. has indicated its advisability. . . . This company has developed the "Comfortractor" with a well sealed cab for farming purposes and at present work is being done on a fan system for bringing fresh, clean air into the cab so that it will not be necessary for the tractor operator to breathe the dust which so often rises in a field.

■ **Alvin H. Baer** has re-joined the Frick Co. as executive assistant after an absence of seven years. He had previously spent more than 30 years with the company, and has served as president of the American Society of Refrigerating Engineers, and the Refrigerating Machinery Association. . . . Morris E. Leeds, founder and president of the Leeds & Northrup Co., assumed last month the position of chairman of the board of directors, and Charles Redding, formerly vice-president in charge of research and engineering, became president. The move will facilitate Mr. Leeds' participation in policy matters. . . . The Spence Engineering Co. has announced purchase of the plant and business of the Rider-Ericsson Engine Corp., which was founded in 1842 by Captain John Ericsson, builder of the warship "Monitor" of civil war fame.

■ **The Air Conditioning** Industries Branch of the Board of Trade of the City of Toronto has recently completed its first year of operations as an association of those engaged in the air conditioning industry. In his annual report, E. C. Williams, retiring chairman, explained that the branch had been formed because of a need for an association embracing those engaged in the industry throughout Canada which would operate to the benefit of both the industry and the public. . . . W. H. Evans was elected to succeed Mr. Williams.

■ **Thirty-nine experts** representing many phases of industrial advertising have been named leaders and co-leaders of 11 industrial advertising clinics to be conducted at the 1939 conference of the National Industrial Advertisers Association at the Hotel New Yorker, New York City, September 20 to 22.

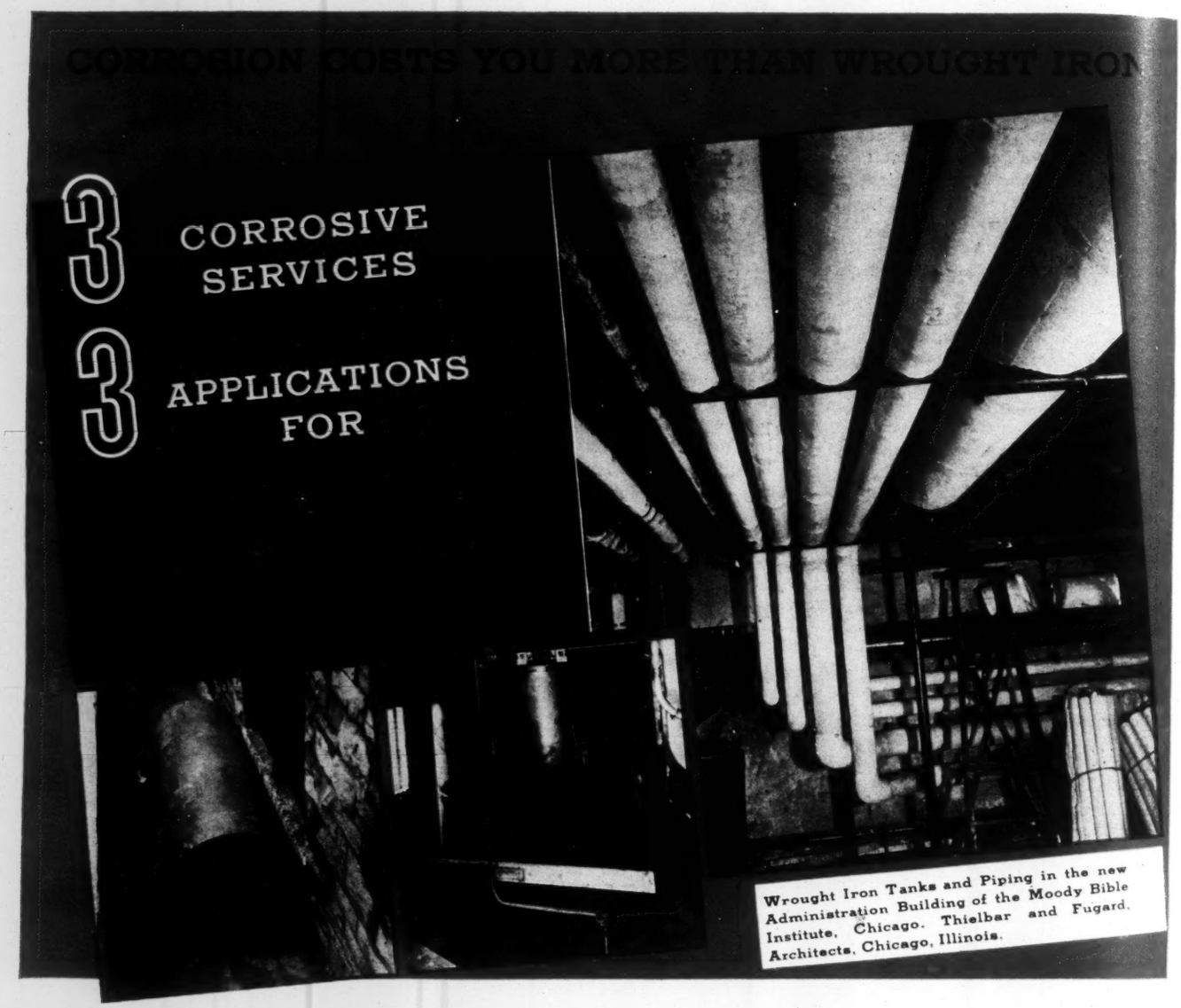
■ **With more actions** being taken on standards than at any other meeting and with a registered attendance of members, committee members and visitors second only to the meeting in New York two years ago, the 42nd annual meeting of the American Society for Testing Materials held in Atlantic City, June 26 to 30, was noteworthy. The program included many round-table get-togethers covering effect of sub-atmospheric temperatures on metals, methods and technique of spectrochemical analysis, freezing and thawing tests, etc. All told, some 116 existing tentative specifications were recommended for adoption as standard, and revisions of standards which had been published previously for consideration are to be referred to letter ballot for adoption in 101 standards. . . . H. H. Morgan, Robert W. Hunt Co., succeeded T. G. Delbridge as ASTM president, and G. E. F. Lundell, National Bureau of Standards, was chosen vice-president to serve with W. M. Barr, Union Pacific Railroad Co., who was elected vice-president last year.

■ **Mechanical Engineering** as a Career is the subject of a 20 p. monograph recently published by the Institute for Research, 537 S. Dearborn St., Chicago, an organization devoted to vocational research. . . . Price of the publication is $1.00 for single copies.

■ **For the first time** in 35 years, except for a short visit in 1932, the Institution of Mechanical Engineers of Great Britain is coming to the United States this fall (September 4-8) to meet with the American Society of Mechanical Engineers in New York. As in 1904, at the time of the St. Louis exposition, a world's fair is to provide the background.

CORROSION COSTS YOU MORE THAN WROUGHT IRON

33 CORROSIVE SERVICES 3 APPLICATIONS FOR



Wrought Iron Tanks and Piping in the new Administration Building of the Moody Bible Institute, Chicago. Thielbar and Fugard, Architects, Chicago, Illinois.

The *engineering approach* to the problem of piping selection was used again in the Moody Bible Institute, Chicago. Experience predicted corrosive conditions in (1) Hot and cold water lines; (2) Cold water storage tank; and (3) Surge tank. Experience recommended Wrought Iron as the best material to combat such corrosion.

The importance of proper piping selection is emphasized by the high cost of premature failure. The wide variety of corrosive services involved, and the wide variety of materials offered to combat them, makes selection an *engineering problem*.

We have helped many architects and engineers to select materials on an engineering basis. We will gladly help you. If you will write, outlining your problem, our Engineering Service Department will determine the probable corrosive conditions; relate these to similar conditions encountered elsewhere; interpret the results in terms of experience gained in over 70 years' contact with corrosion problems, and make recommendations—backed by actual service records. Ask, too, for a copy of the bulletin, "Wrought Iron in Piping Systems." May we work with you?

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GENUINE WROUGHT IRON
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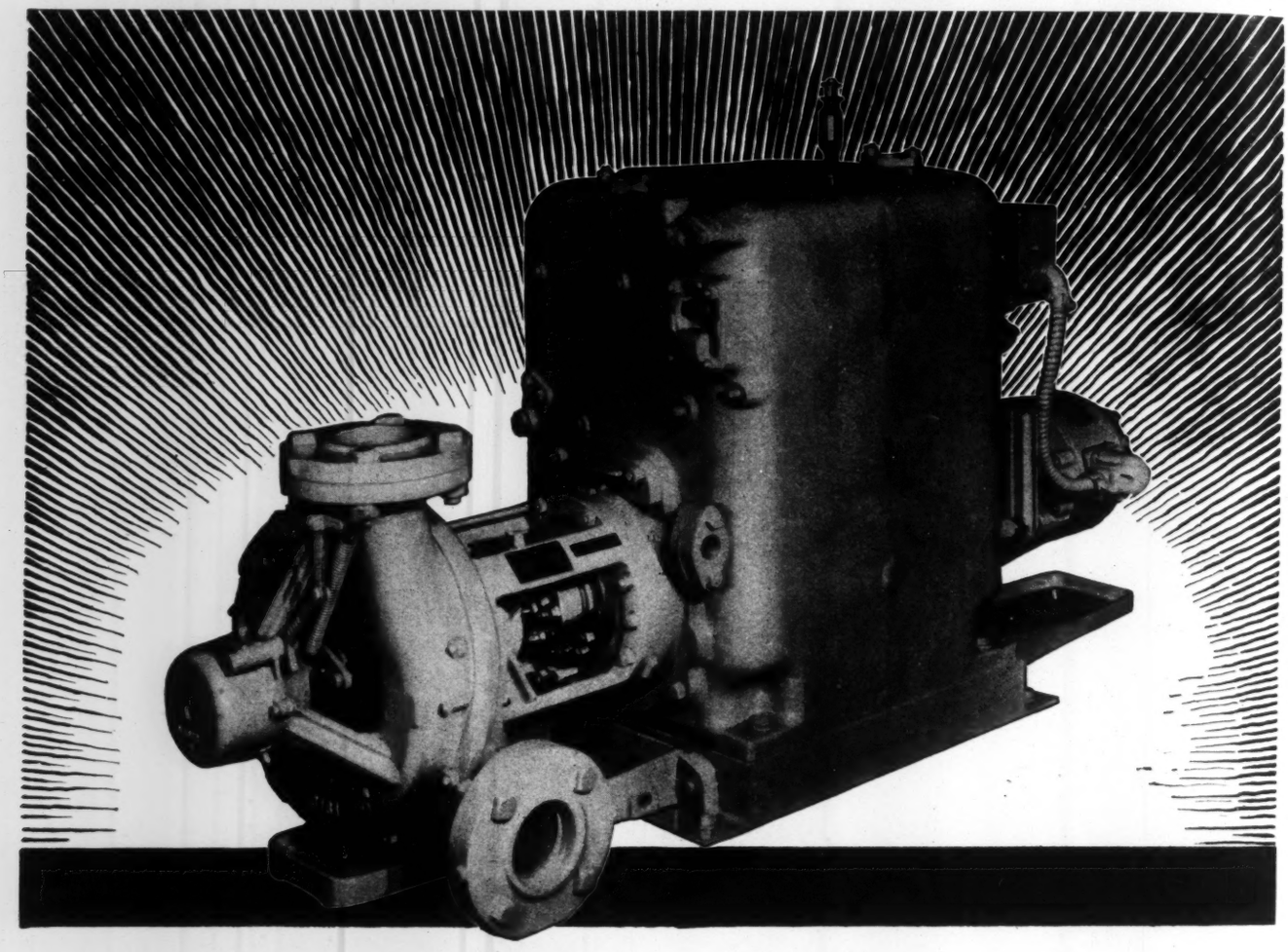
Safety!

THE NASH VAPOR TURBINE HEATING PUMP IS NOT STOPPED BY ELECTRIC CURRENT FAILURE



For service in greenhouses, hospitals, schools, theatres, and public buildings, where heat failure is a serious matter, the Nash Vapor Turbine Return Line Vacuum Heating pump should be specified, for this pump operates perfectly without electric current.

This is because the motive power is a special turbine operating on steam direct from the heating system. Steam used to drive the Vapor Turbine is returned to the system for heating, with little heat loss.

This pump insures constant heat in the system, it eliminates electric current cost, and it promotes high efficiency in the heating system. Heating efficiency is due to the fact that this pump can operate continuously with economy. Continuous operation means uniform circulation, and uniform conditions in the system mean steam saving.

This pump has but one moving part, no internal wearing parts, and requires no internal lubrication. Bulletin D-246 tells all about it, and it is free on request.

THE NASH ENGINEERING COMPANY

SOUTH NORWALK, CONNECTICUT, U. S. A.

Heating · Piping and Air Conditioning

Vol. 11, No. 8 • *August, 1939*

Piping Design and Installation at the Irvin Works of Carnegie-Illinois Steel

PIPING systems for over 20 different liquids and gases are in service at the new Irvin Works of the Carnegie-Illinois Steel Corp. in the Pittsburgh district. Coördinated effort during installation of this piping was essential, as so many different classes of work were going on at the same time during the erection of the mill.

For example, in the hot mill building a 12 in. welded steam line runs for 2232 ft through the trusses, carrying steam at 520 F and 200 lb pressure. Roller bases were attached to the bottom chord members before the trusses were raised into position. Pipe work was all done from one position, jacking the line through the trusses as they were placed, by means of winches, using a 1½ hp motor.

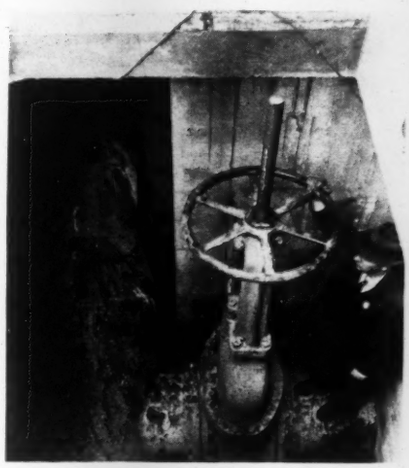
Seamless steel pipe, scarfed for welding, was delivered in 40 ft lengths to the pipe station. Here it was raised 51 ft to the truss line and arc welded to the advancing main. After the straight line had been completed, a crew installed expansion loops every 250 ft, cutting out a section of the straight pipe and welding the bends in position. The steam main was anchored approximately every 250 ft, between the expansion loops, the trusses being specially braced to withstand the strain.

A 20 in. coke oven gas line was installed through the center of the trusses in the annealing building. Van stone flanged pipe was used. The pipe was furnished in 20 ft lengths, flanged at one end, and was welded at the erecting point into 40 ft lengths. These sections were erected with a 65 ft boom crane. Two expansion joints were installed in this line.

Glass Tubing Inside Copper Tubing

Acid handling presented an interesting problem in the hydrogen evolution laboratory. Here chemically pure hydrochloric acid must be delivered from the carboys to the benches and hoods. Piping is ½ in. glass tubing inside 1½ in. copper tubing, the space between the glass and copper tubing being filled with beeswax. All glass tubing was delivered in straight lengths, bent on the job as required, and fused in place. As the work progressed the copper tubing was slit, sprung in place around the glass and then brazed, after which the space between the two tubes was filled with melted beeswax.

All palm oil and paraffin oil lines are installed with



companion steam lines, both lines being insulated and wrapped together. Steam, condensate, palm oil, and paraffin oil are insulated with sponge felt, 2 in. of insulation being used for pipes 2 in. and under, 2½ in. for those between 2 and 4 in. diameter, and 3 in. of insulation for pipes 4 in. or larger. Similar exterior lines have at least ½ in. additional insulation. Cold water line insulation is 1½ in. thick wool felt to prevent freezing, while hot water pipes carry 1 in. of insulating felt.

To facilitate identification as well as to improve appearance and simplify maintenance, all piping is painted according to the accompanying schedule. All piping is painted with the base color, and stencil lettering is applied at locations adjacent to valves. In long runs of pipe, a strip or ring 4 in. wide is painted around the pipe approximately every 24 ft.

The piping at the new Irvin Works of the Carnegie-Illinois Steel Corp. includes systems for more than 20 different liquids and gases. Details of the installation and design are described here, including the method of identifying the various services, use of explosion heads as a safety measure, piping materials and connections, etc. . . . As constant functioning of the slab heating furnaces at the hot mill is vital, four alternate systems of water supply for the furnace skid cooling water are available

Orifice Plugs and Explosion Heads for Safety

Numerous large unit heaters are installed throughout the plant. These 1,000,000 Btu heaters are connected to the main 200 lb, 520 F steam lines by 2 in. pipes, a valve being installed on the line at the main. However, as a safety factor, a ⅜ in. orifice plug is installed between the main line valve and the heater so as to limit the steam flow to the maximum which the heater can use. Operating at capacity, each of the large unit heaters takes approximately 1000 lb per hr of steam.

As an extra precaution, 35 lb explosion heads have been installed on the low pressure side at all reducing stations between the main high pressure steam lines and offices, wash and locker rooms, or points wherever men may be working in confined quarters. The main line carries approximately 200 lb steam pressure, which for these heating purposes is reduced to 10 lb through reducing valves, protected by 15 lb safety valves.

As a special safety measure, a 2300 lb explosion head is contemplated in connection with the accumulator used in connection with the descaling system in the 80 in. continuous hot strip mill. The nozzle operating pressure at the descaling stations is 1350 lb per sq in. The accumulator is of laminated welded steel plate construction, 12 ply, and is designed for a pressure of 2000 lb per sq in. Pump shut-off pressure is 1600 lb.

Schedule for Identification of Piping

Type	Base Color	Stencil or Stripe	Stencil
Hydraulic (descaling) water, 1500 lb	Yellow	Orange	DESCALING
Hydraulic (squeezer) water, 600 lb	Yellow	Green	SQUEEZER
Raw water, 85 lb	Green	Red	.....
Filtered water, 75 lb	Green	White	.....
City water, 60 lb	Green	Purple	.....
Soda water	Green	Black	.....
Sprinkler and foamite	Red	.....	.....
Acid showers	Blue	.....	.....
Compressed air, 110 lb	Green	Orange	AIR
Coke oven gas, 6-8 lb	Yellow	White	C.O. GAS
Natural gas, 9 lb, Main, 10-30 lb	Yellow	Purple	NAT. GAS
Mixed gas	Yellow	Black	MIX. GAS
Deoxidizing gas	Yellow	Green	DX. GAS
Hydraulic roll balance, 3000-3500 lb	Orange	.....	.....
Palm oil and palm oil coolant	Orange	Green	P.O.
Paraffin oil	Orange	Yellow	PARAFFIN
Kerosene	Orange	White	KERO.
Pinion stand oil	Orange	Purple	PINION
Reduction gear oil	Orange	Brown	RED. GEAR
Motor and M. G. set oil	Orange	Black	MOTOR
Bearing oil	Orange	Gray	.....
Acid (sulphuric or muriatic)	Yellow	.....	ACID
Steam quenching	Red	Blue	.....
Acetylene	Yellow	Gray	.....
Oxygen	Yellow	Brown	.....

Skid Cooling Water

Constant functioning of the slab heating furnaces at the hot mill is vital to the operation of the mill. As the skid pipes in these furnaces are all water cooled, every precaution must be taken to assure an ample and continuous supply of water, for furnace temperatures run up to about 2400 F. Standard practice is to take the skid cooling water from the hot mill motor room cooling system. However, four other alternate systems of water supply are available in case of need: (1) Should the motor room cooling system fail, a special raw water supply is automatically turned into the skid pipes. (2) A direct line has been installed between the furnaces and the service water lines at the pump room. (3) If necessary, the furnace cooling water can be recirculated through the skid pipes by means of a service pump. (4) A tie-up has been installed whereby in emergencies, filtered water from the plant reservoir can be circulated through the skid cooling system.

All of these plans have been tested, and it is interesting to note that in nearly a year's operation, at no time have the furnaces been shut down because of faulty skid cooling. All lines to the furnaces are equipped with check valves so that back flow of the cooling water is prevented when any of the alternate cooling systems are put into operation.

Miscellaneous Piping Details

Various types of valves are used at the Irvin Works. Ordinarily all large valves are steel body units with stainless steel seats and discs of dissimilar metal. Gas valves are all iron, many of them being lubricated cocks. Most of the smaller valves throughout the plant have bronze bodies, and the small globe valves are ordinarily of the plug type. Steam traps are all made of forged steel and are equipped with stainless steel buckets.

As a safety measure, high pressure descaling pumps are connected to their lines through two male flanges. This permits a loose ring to be fitted over the gasket at the joint, under the flange bolts. Thus, if the gasket should blow out, the loose ring will function as a baffle and break up the stream, reducing its velocity below the danger point.

Screwed flanges have been eliminated wherever possible, about the only exception being in the case of drinking water lines, where steel screwed flanges are used on lines from 3 to 6 in. in diameter. In place of screwed flanges, van stone flanges, weld neck flanges, or slip-on flanges have been used.

Practically all bends were made outside the plant and fitted into position on the job by welding. Random bends were frequently used, as pipe lengths were easily adjusted by means of a cutting torch. Tunnel piping was installed whenever possible before the roof was placed and it was a constant race between the piping crew and the other construction crews, requiring careful planning on the part of the piping engineers and close coördination on the part of the general contractors to keep all gangs working at full speed without conflict.

[Photographs courtesy Power Piping Div. of Blaw-Knox Co.]

Sales Development Keynotes Meeting of National District Heating Association

SALES development was the keynote of the 30th annual convention of the National District Heating Association held at the Pennsylvania Hotel, New York City, June 27 to 30. The meeting opened Tuesday morning, and the afternoon session was given over to the sales development committee. This committee's report consisted of several inspirational addresses. The speakers were F. L. Carlyle, chairman of the board, Consolidated Edison Co.; Alex Dow, president, The Detroit Edison Co.; F. B. Bryans, executive vice-president, Philadelphia Electric Co.; Miss Clara H. Zilessen, advertising manager, Philadelphia Electric Co.; and Merle Crowell, in charge of public relations, Rockefeller Center. Many utility executives were present to hear leaders of the industry talk, and more than 300 were in attendance at this meeting.

Wednesday, Thursday, and Friday sessions were given over to technical committees, the Wednesday session being devoted to reports of the commercial relations and research committees. The Study of Heating Economies, a paper by Earle Shultz and J. C. Butler, was a progress report of savings obtainable by the use of temperature controls and other heat saving devices. Continuation of this report will be watched with interest. A paper on steam compressors, by G. H. Tuttle, gave a description of a method of increasing steam pressure for process work and created considerable discussion by manufacturers and delegates.

The Thursday session was opened by a report of the steam station engineering committee, which was unusual in that it gave moving pictures of fuel beds under various fuel bed conditions within boilers. The feature of this is that after development of methods of taking such moving pictures, studies will without doubt show how furnaces and stokers can be improved in design and operation to give higher efficiency. The paper by E. P. Durfee on Safety Pays Dividends was well received. Mr. Durfee showed that actual savings have resulted from the adoption of safety measures in power plants. The distribution committee report gave a summary of the practices of operating companies in the details and methods of construction.

Highlights of the year 1938 as summarized in the report of the operating statistics committee included (1) the greatest decrease in sales volume and revenue for at least 10 years, (2) universally warmer weather during the heating season throughout the entire territory, (3) a definite decrease in building occupancy, particularly in commercial hotels, (4) a mixed trend in the price of fuels, coal and hogged fuel (saw-mill waste) advancing, while fuel oil declined, (5) a decrease in the number of steam employees and consequent reduction in the total cost of labor, and (6) a slight but significant loss in connected load. The meters and accessories committee report gave a comprehensive compilation of meter practices and construction by member companies and meter manufacturing companies.

Much interest was shown in the exhibits of 42 manufacturers of equipment.

William J. Baldwin and Colonel Davis S. Boyden were voted honorary members of the Association. Colonel Boyden is a past president of the American Society of Heating and Ventilating Engineers, and both men are past presidents of the National District Heating Association.

The following were elected to office: G. D. Winans, president; F. L. Witsell, first vice-president; L. S. Phillips, second vice-president; J. M. Arthur, third vice-president; executive committee—J. R. McCausland, D. C. Turnbull, and R. M. Nee.

Static Electricity in Sawdust Duct Eliminated by Use of Water Sprays

By Alf H. Hofberg*

SAWDUST conveying systems handling large volumes at high velocities present an explosion hazard due to several causes. In our efforts to prevent any such damaging accidents, we have catalogued all the possible causes that came to mind, among which is the possible existence of high potential electrostatic charges. We decided on a series of tests to confirm the presence of any dangerous charges.

Apparently, electrically charged particles are not dangerous until the potential reaches the neighborhood of 500 volts or more. An electrostatic voltmeter with variable ranges was prepared and an electrode made with an insulated handle. Holes were drilled in the sawdust ducts at various points. These holes gave plenty of

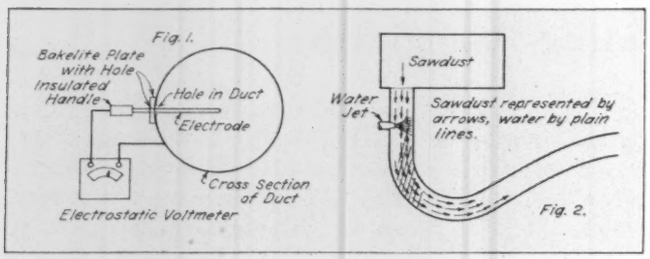


Fig. 1—Method of taking readings with electrostatic voltmeter. If such an instrument is not available, a gold leaf electroscope will indicate the presence or absence of a charge. . . . . Fig. 2—The three water jets each have a pet cock so any or all may be used. A solenoid valve in the water feed line opens when the blower is in operation and closes when the blower shuts down

clearance to the electrode and were covered with bakelite plates which also had holes in them that would just clear the electrode.

The meter and electrode were connected as shown in Fig. 1 and we immediately got definite evidence of dangerously high electrical charges. Tests were run over a period of two or three weeks, and we were lucky in having a great variety of weather during the test period. On warm, humid days the voltage was very low, not over 50 or 75 volts, or entirely non-existent. However, on colder, dry days the charge would go as high as 1500 or 2000 volts.

The test showed two things of importance. The static electricity in the duct was at times dangerously high and the amount definitely varied with the humidity. In fact, high humidity was always accompanied by a disappearance of electrical charges. Therefore, we concluded that we could make our system safe—at least from explosion due to electrical discharges—by introducing moisture.

In all our tests we found that it was only in the long ducts where the velocity of the air driving the sawdust was high—7000 fpm—that noticeable charges existed.

We introduce the moisture at the start of the duct. This particular duct is approximately 1200 ft long. The water jets are mounted as indicated in Fig. 2. There are three jets of a self-cleaning type, all set at the same level and within 3 in. of each other. Each jet passes about 1 qt per min at 30 lb pressure. The water feed line to the jet is controlled by a solenoid valve so that the water is turned on automatically when the blower is started and also turned off when the blower is stopped. There are pet cocks to each jet so that either one, two or three jets can be on at any one time. Only in extremely dry, cold weather do we find it necessary to use all three jets. With the jets in operation, there are no detectable electrical charges in the ducts.

No trouble has been experienced by the sawdust becoming wet, which might cause packing in the ducts.

The electrical measurements and also humidity checks were made in the duct about 900 ft from the point at which the water is introduced. With the water off, on a cold, dry day when the voltage was in the neighborhood of 1500 volts, it dropped to zero volts in 20 sec after the water was turned on. This checked with the calculated time it would take the sawdust to travel 900 ft in the duct at the pressure and volume of our blower, and with the conditions under which the air is admitted in puffs.

If no electrostatic voltmeter is available, a satisfactory substitute can be made in the form of a gold leaf electroscope for a very nominal sum. Although the electroscope will not tell accurately the amount of electricity present, it will indicate the presence of any voltage.

Static Electricity in Blower Systems

The Underwriters' Laboratories, Inc., 207 E. Ohio St., Chicago, recently issued a bulletin on Generation of Static Electricity in Blower Systems. From the tests reported it appears that air practically free from solid or liquid particles is not perceptibly electrified under conditions ordinarily obtaining in blower systems. If dust or lint is present in appreciable amounts, static charges of a magnitude capable of causing ignition by spark discharges may be generated. Voltages actually observed ranged from 100 to over 10,000. Within certain limits the static charge increased in magnitude with increase in the concentration of dust or lint, fineness of particles, velocity of air stream, and decrease in humidity. While the subject of protection against static in blower systems is outside the bulletin's scope, it is evident that blower systems for dust, lint or vapor removal should be electrically grounded to prevent accumulation of dangerous static charges, the bulletin states.

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The Psychrometric Chart

Its Application and Theory

By William Goodman*

FOR the conditions illustrated in Fig. 2, shown on p. 358 of the June issue, the total heat carried away by the air-vapor mixture leaving the chamber is equal to the sum of the sensible heat, Q†, which raises the temperature of the air and the heat of the moisture, $W_o h_o$. The ratio of the sensible to the total heat added to the air may be defined by the following equation, in which q_s is called the *sensible heat ratio*.

$$q_s = \frac{Q}{Q + W_o h_o} \quad \ldots\ldots [6]$$

This equation is valid only as long as the moisture added to the air is in the form of saturated or superheated steam. The case in which moisture is added in liquid form is discussed later.

A table similar to Table 1 on p. 422 of the July issue could be provided to give the *ratio angles* corresponding to the various sensible heat ratios. For a given problem, the ratio angle would be the same regardless of whether it was determined by means of the sensible heat ratio or the *moisture ratio*. The following equation, developed in Appendix 2, shows the relationship between the sensible heat ratio and the moisture ratio.

$$q = \frac{h_{g2}}{1 - q_s} \quad \ldots\ldots [7]$$

For moderate temperatures, the following approximate equation may be used.

$$q = \frac{1061}{1 - q_s} \quad \ldots\ldots [7a]$$

When the moisture added to a conditioned space is in the form of steam, as is the case in comfort conditioning, the computation of the sensible heat ratio is simple. On the other hand, if the moisture is in the form of liquid, then, even though the computation is simple, care must be exercised in order to arrive at a correct numerical value for the sensible heat ratio; an error is easily made under these conditions. The difficulty arises from the fact that whereas the moisture is added to the conditioned space in the form of liquid, it is carried out of the conditioned space in the form of vapor by the air stream. But the heat needed to vaporize the liquid can be obtained only from the sensible heat gain of the room or from the cooling of the air supply. Consequently, when computing the sensible heat ratio, the latent heat required to vaporize the water must be deducted from the sensible heat gain. When the moisture entering the chamber is in the form of either saturated or superheated steam, equation 6 can be used. However, if the steam is unsaturated or if water alone is supplied, the following equation must be used. The derivation of this equation is obvious.

$$q_s = \frac{Q - W_o (h_{g2} - h_o)}{Q - W_o (h_{g2} - h_o) + W_o h_{g2}} \quad \ldots\ldots [8]$$

$$= \frac{Q - W_o (h_{g2} - h_o)}{Q + W_o h_o} \quad \ldots\ldots [9]$$

In deriving [8] it is important to remember that it represents not the ratio of the sensible to the total heat gain of the *room*, but the ratio of the sensible to the total heat *added to the air stream*. Hence the term $W_o h_{g2}$ appears in the denominator of [8] because the moisture is carried away in the form of vapor, even though it is supplied in the form of liquid. When all of the moisture supplied is in the form of saturated steam or superheated steam, $h_{g2} = h_o$, and equation 9 reduces to equation 6.

Where the moisture added to the air stream is in the form of vapor, the use of the sensible heat ratio requires no special consideration. But, when the moisture added is in the form of unsaturated steam or in the form of liquid there is a possibility of making an error unless care is exercised — especially by those who use the method only occasionally. On the other hand, the use of the moisture ratio involves no special consideration regardless of whether the moisture added to the air is in the form of superheated steam, unsaturated steam, or liquid water. For this reason, there is no possible chance of error if the concept of moisture ratio is used. The advantage of using the moisture ratio is that it is ap-

In addition to the elementary uses of the psychrometric chart—such as showing the relationships between dry bulb, wet bulb, and dew point temperatures, and relative humidity—it provides a simple and enlightening means of analyzing many complex problems in comfort and process air conditioning, including drying. . . . New psychrometric charts covering the low, middle and high temperature ranges, with saturation curves for various barometric pressures, are presented here. The general theory of the psychrometric chart is discussed in detail, and the methods of solving problems by the use of charts are described. . . . These new charts facilitate solution of most problems and lead to clearer understanding of the fundamentals

*The Trane Co. Member of Board of Consulting and Contributing Editors.

†See table of symbols, p. 358, June issue.

Part 3. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, June, 1939, pp. 357-360, and Part 2 in July, pp. 421-424.

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plicable under all conditions regardless of the form in which the moisture is added or removed during the conditioning process.

Because of the reasons stated in the preceding paragraph, the use of the moisture ratio for computing the ratio angle is preferable to the use of the sensible heat ratio. This is true even for problems in summer comfort conditioning where the sensible heat ratio has found its widest use. For such problems, the use of the sensible heat ratio seems to have some psychological advantage because its maximum value is one, and hence it gives the proportion of the total heat gain of a room that is in the form of sensible heat. Nevertheless, even for these problems there is no real advantage in using the sensible heat ratio instead of the more basic and logical moisture ratio. Because the concept of the moisture ratio is general, and because it can be applied to all types of problems, its use is preferable to the use of the sensible heat ratio.

The Energy Equation

The method of analysis which carries the somewhat formidable name of the "energy equation method" is illustrated by Fig. 2 and equation 1 (see Part 1 published in June). This represents a simple case which was purposely chosen in order to show that an analysis by means of the energy equation method consists of nothing

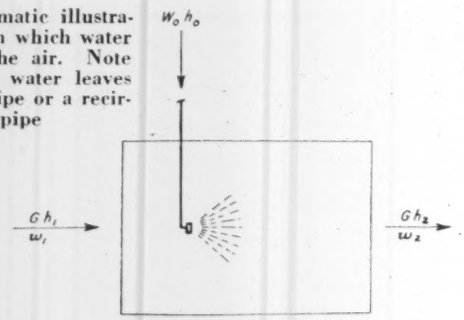


Fig. 7 — Diagrammatic illustration of chamber in which water is sprayed into the air. Note that none of the water leaves through a drain pipe or a recirculating pipe

more than a statement that the energy flowing into a room or apparatus must equal the flow of energy out of it after steady flow conditions have been attained. By analyzing a problem in this way and setting up the equations in the manner illustrated by equation 1, the most complex problems in air conditioning can be readily analyzed. This method will be followed throughout the balance of this discussion.

There is nothing very startling about the basic concept or application of the energy equation. Yet because it lends itself so admirably to the analysis of various thermal processes, it has proved a tremendously important weapon in attacking many types of problems. As Professor Keenan[8] has written of this method: "It is hardly more than 10 years since the beginning of a revolution in the exposition of thermodynamics of fluid flow. This revolution consisted not in the discovery of a new principle but in the recognition of the importance of an old one. . . . It has brought precision and simplicity into the exposition of engineering thermodynamics. . . . It has eliminated much complex and fruitless reasoning. . . . "

[8]J. H. Keenan, A Steam Chart for Second-Law Analysis, *Mechanical Engineering*, March, 1932, p. 195.

Mixtures of Air and Water

Today the fact is quite well understood that if water at the same temperature as the wet bulb temperature of the air is sprayed into a stream of air, the water will evaporate into the air. The wet bulb temperature of the air will not change, but its dry bulb temperature and absolute humidity will change. On the other hand, the results obtained are not quite so well understood for the case where the temperature of the water *differs* from the wet bulb temperature of the air. The result of spraying water at any temperature into air will be discussed in this section.

In this section the assumption will be made that all of the water sprayed into the air either evaporates into the air stream or remains in suspension in the air. In other words, the water either evaporates into the air or, if it is not evaporated, it is entrained and carried away in suspension by the air stream. For the purpose of this discussion, none of the water that is sprayed leaves the apparatus by means of a drain pipe or a water recirculating pipe. The case where the water leaves the system either through a drain pipe or a recirculating pipe is discussed in a later section.

Referring to Fig. 7, the heat balance or energy equation for this case may be written

$$Gh_1 + W_o h_o = Gh_2$$

$$G\,(h_2 - h_1) = W_o h_o \quad \ldots\ldots [10]$$

Also

$$Gw_1 + W_o = Gw_2$$

$$G\,(w_2 - w_1) = W_o \quad \ldots\ldots [11]$$

Dividing [10] by [11],

$$\frac{h_2 - h_1}{w_2 - w_1} = h_o \quad \ldots\ldots [12]$$

Inasmuch as the left-hand members of both equations 4 and 12 are the same,

$$q = h_o \quad \ldots\ldots [13]$$

In other words, the moisture ratio is equal to the initial enthalpy of the water supplied to the chamber. Inasmuch as the enthalpy of the water supplied to the chamber is constant and independent of any changes that may take place inside the chamber, the final condition of the air leaving the chamber must lie on a ratio line. The slope of this ratio line depends only upon the enthalpy of the spray water. As will be shown later, this is one of the few cases where the ratio line actually represents the change in the condition of the air as it flows through the chamber in which the moisture is added to the air.

Example 6: Air at an initial dry bulb temperature of 85 deg and a dew point temperature of 40 deg is passed through a water spray. The initial temperature of the water is 120 deg. The quantity of water evaporated into the air amounts to 1.5 lb per min. The quantity of air passing through the spray is 375 lb per min. The barometric pressure of the air is 26 in. Find the final dry bulb and dew point temperatures of the air.

Solution: For water at 120 deg, $q = h_o = 88$ Btu (from Keenan and Keyes steam tables). Referring to Table 1 (p. 422, July issue) for $q = 88$ Btu, $\theta = -55°\text{-}46'$. Draw the ratio line through the point on the psychrometric chart (Chart

A, p. 359, June issue) located at 85 deg dry bulb and 40 deg dew point temperature.

$$\frac{W_o}{G} = \frac{1.5}{375} = 0.0040 \text{ lb of water evaporated per lb of air.}$$

From the psychrometric chart, $w_1 = 0.00599$.

Using equation 2, $w_2 = 0.00599 + 0.004 = 0.00999$ lb, final absolute humidity of the air.

The point representing the final condition of the air is located at the intersection of the horizontal line of absolute humidity of 0.00999 lb and the ratio line. The final dry bulb temperature of the air leaving the chamber will be 68.5 deg, and the final dew point temperature 53.4 deg.

The coldest water that can be sprayed into air is 32 deg and the warmest—at normal barometric pressure—is 212 deg. The enthalpy of the water corresponding to these temperatures is 0 Btu and 180 Btu respectively. Referring to Table 1 for $q = 0$, $\theta = -53° -25'$, and for $q = 180$, $\theta = -58° -21'$. Drawing the lines corresponding to these two enthalpies on the psychrometric chart of Fig. 8, it is evident that the ratio lines for all water temperatures between 32 and 212 deg must lie in a very narrow sector on the psychrometric chart. The angle between the two extreme ratio lines for water at 32 deg and 212 deg is small. Hence, where the water sprayed into the air is entirely evaporated into the air stream, the change in the final condition of the air is not affected much by the temperature of the water. This conclusion is valid only for the case where none of the water overflows through the drain pipe or is carried away through a recirculating pipe. The case in which some of the water leaves the conditioning apparatus through a drain pipe or through a recirculating pipe is discussed in detail in a later section.

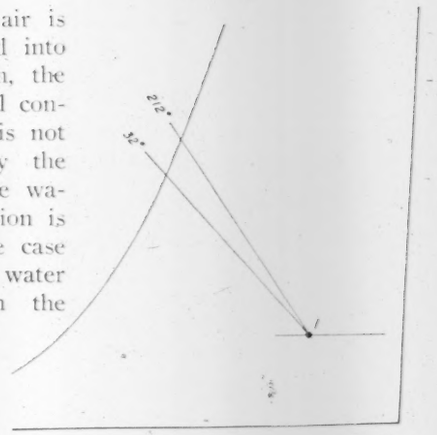


Fig. 8—Ratio lines corresponding to water at 32 deg and 212 deg. Note that even though the range of water temperatures is fairly large, all of the ratio lines for water fall in a small sector of the psychrometric chart. In this illustration the angle between the lines has been exaggerated for the sake of clarity

[To be continued]

Decorating Air Conditioning Grilles

HOW to decorate large air conditioning grilles in the ceiling of a handsome banking room became a problem at the Hibernia building in New Orleans, managed by Waguespack, Pratt & Hogan, Inc., according to *Buildings and Building Management*. The ground floor of the building is occupied by the Hibernia bank, with elaborate banking quarters. When this room, along with the rest of the building, was air conditioned, the grilles were installed in the expensively decorated ceiling, and, with all ordinary decorating treatment, appeared as dead black holes.

F. Poche Waguespack states the problem and its solution thus: "The various standard colors in which grilles could be obtained were tried, but they made an appearance such as is shown in the right of the photograph—more or less black holes in the ceiling. The spraying of the grilles the color of the field was attempted, but inasmuch as the grilles have directional fins, reflection of the light on the fins which faced in different directions gave a mottled or streaked effect. Running stencils were tried, but this gave the same streaked appearance. By experimentation we found that by using a stencil design, the different directions were concealed. This is shown on the left-hand side of the photograph."

A similar problem was encountered in transoms over office doors. Mr. Waguespack explains: "Air conditioning was installed in all of the upper floors, as has been done in other instances, by placing the ducts in the corridor. In so doing, transoms were blanked out completely on the corridor side, but remained facing on the office side. Painting of the transoms left a rather dead look which was not in keeping with the appearance of the offices. A solution was finally arrived at by placing tinfoil-covered building board such as is used for insulation purposes about an inch back of the transoms. This permitted the light to be reflected, and gave the transoms the appearance they would have had without being obstructed."

By using a stencil design as shown at the left, the appearance of the grilles in an expensively decorated ceiling was improved. The right view shows how they looked before the design was applied

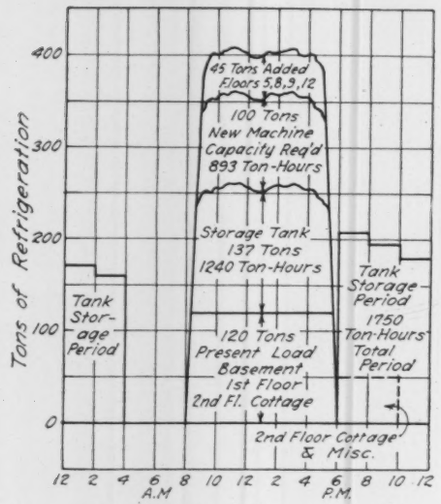


Fig. 1 - Typical Summer Day Heat Load

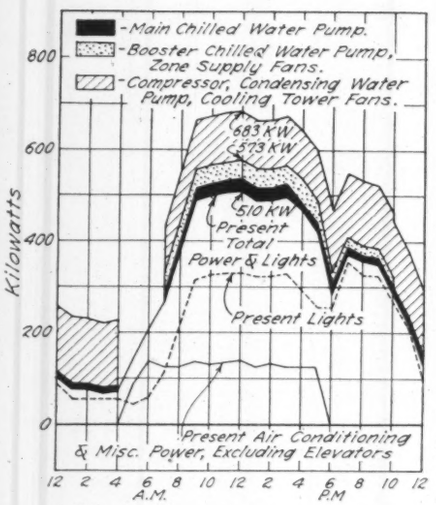


Fig. 2 - Typical Summer Day Electric Load

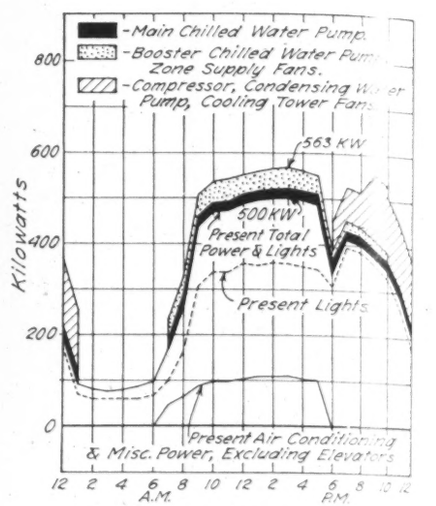


Fig. 3 - Typical October Day Electric Load

How Storage Principle Is Applied in Air Conditioning an Office Building

By D. C. Pfeiffer*

THE main office building of the Dallas Power & Light Co. in Dallas, Texas, was built in 1930 and consists of 18 floors. The second through the 15th floor is devoted to office space. The building is of welded steel skeleton construction, with concrete and tile floors and brick walls. When the structure was built, there was provided an air conditioning system for serving the first floor, the auditorium and demonstration cottage on the second floor, and part of the basement. The equipment consists of a 110 ton centrifugal compressor, together with necessary condensing water and chilled water pumps and fans. The conditioning unit is a single stage, spray type dehumidifier. Total connected motor load is 226 hp.

Early in 1938, it was decided to provide the office portion of the building with a storage type air conditioning system. Fig. 1 gives a typical daily refrigerating load curve of the building showing the original system, together with the additional load of the office portion of the building. Estimates of the heat gains in the building indicate that approximately 282 tons of refrigeration are required for the second to the 16th floors inclusive. The 17th floor auditorium and four unoccupied floors were not equipped for air conditioning at this time, but the refrigerating plant was made large enough so that in the future these floors could be added with a minimum amount of modification. The estimated load of the part of the office building actually conditioned is 237 tons.

It was decided to use the sweet water type of storage system. Investigation of the basement showed a space approximately 56 ft by 20 ft to be available. A storage tank was installed in this location with a capacity of approximately 13,500 cu ft of water. The effective capacity of the tank was based upon cooling the water to 36 F and the conditioning units were designed for a leaving water temperature of 58 F. On this basis each pound of water when warmed from 36 to 58 F in the air conditioning system will extract 22 Btu, and allowing 62.4 lb of water per cu ft, the total thermal capacity of the tank is 18,500,000 Btu. On a ton-hour basis this will require 1550 ton-hours to cool the water from 58 to 36 F. Allowing 20 per cent for the tank "losses" (incomplete water displacement and heat pick-up) there will be available 1240 ton-hours for use in air conditioning. Using a 90 per cent factor for diversity, this tank has been given a normal 10 hr rating of 137 tons equivalent machine capacity.

With the building load estimated at 237 tons and allowing 90 per cent diversity, there will be required 2133 ton-hours of refrigeration for a 10 hr day. Subtracting the 1240 ton-hour capacity of the tank, a balance of 893 ton-hours must be supplied by the compressors during the day. The compressor equipment was selected so that one of the machines would normally carry this load and the second machine would be used at night for cooling the tank.

It is assumed that the full 1550 ton-hours will be required for the night cooling of the tank with an additional 200 ton-hours representing miscellaneous night load. This gives a total ton-hour input to the tank for night cooling of 1750.

*Sales Dept., Dallas Power & Light Co.

Fig. 2 shows a typical 24 hr electric load curve of the building for a typical summer day, the air conditioning operation being shown as superimposed upon the normal building load. Fig. 3 shows a typical October day load curve showing the air conditioning system operating as 100 per cent storage. Fig. 4 is a diagrammatic arrangement of the chilled water piping showing the location of pumps, valves, thermostats and controls necessary for the operation of the storage system. The added refrigerating equipment consists of two 8 in. by 8 in. 360 rpm ammonia compressors complete with spray type chilled water cooler. A forced draft cooling tower is situated on the roof of the building. The building is conditioned with closed coil type conditioning units on the various floors. A chilled water piping system furnishes water to the units at 50 F, returning at 58 F. The storage system is known as a semi-storage type in which the refrigerating equipment is operated both day and night. During the day the refrigerating effect of the cooled water is utilized to assist the compressors to cool the building. This is accomplished by mixing chilled water from the tank with part of the returned water from the system, to maintain the desired water temperature.

Normal Day Operation

Normal day operation is as follows (see Fig. 4):

Main chilled water pump *2* circulates water through the cooler to the supply riser, through the conditioning units, to the return line and back to the suction side of this pump. A thermostat in the supply riser operates a thermostatically controlled valve *A*. When the requirements for refrigeration exceed the capacity of the cooler, the temperature of the water rises and the thermostat controls valve *A* in such a manner as to allow water from injection pump *3* to enter the closed loop. The injection pump operates only at such times as the thermostatically controlled valve *A* is open.

The water that is injected into the closed loop under pressure displaces an equal amount of water in the surge tank, which amount of water is piped back to the warm end of the storage tank by means of the overflow line shown on the drawing. The system is operated in such manner that during a normal day the entire quantity of water in the storage tank is pumped into the system so that at the end of the day the temperature in the storage tank is 58 F. The baffling arrangement of this tank is such that the water travels around a series of vertical baffles so that the temperature of the water drawn from the cold end of the tank is not affected by the 58 F water returning from the system. An allowance of 20 per cent was made in the determination of the size of this tank to allow for incomplete stratification of the water. The capacity of the injection pump *3* was selected so that during a normal 10 hr day it pumps the volume of water of the tank.

Night Operation for Cooling Tank

At a predetermined time in the evening, when the cooling requirements for the building are over, the main chilled water pump *2* is connected by closing valves *1*, *4* and *5*, and opening *7* and *11*, so that water may be pumped from the warm end of the water storage tank through either of the coolers to the cold water end of the tank. This continues until the entire water temperature in the storage tank has been reduced to 36 F.

The main chilled water pump *2* is equipped with a variable speed, wound rotor induction motor, which permits operating the pump at various rates of flow so that the time of cooling the tank can be varied according to the judgment of the operator.

Night Operation of Building

If at any time it is desired to furnish air conditioning on any of the floors during night hours, this can be accomplished by operating the injection pump *3* at the same time the main circulating pump is used for cooling water in the storage tank. Under these conditions the thermostatically controlled throttle valve *B* permits pump *3* to receive its supply of water either from the return line to the warm end or from the low temperature end of the storage tank, depending upon the position of valve *B*. Each of the motor operated control valves determining the supply of water to its conditioning unit is so interlocked with the motor circuit of the fan that this valve is closed unless the fan is in operation. If, for example, the 14th floor desires air conditioning in the evening, the fan unit on the 14th floor can be started from a push-button station in the basement and injection pump *3* operated as described. Under these conditions, with valve *5* closed, all of the water which is pumped into the system must return through the overflow line. It is apparent that the number of units which can be operated under this condition is limited by the capacity of pump *3*.

Instruments and Controls

One of the outstanding features of this air conditioning system is the instrument and control board. In order to determine accurately the thermal effectiveness of the storage tank, a means has been provided for metering the Btu output of one of the coolers, using a flow meter having two temperature pens on the same chart. These temperatures are designated by t_1 and t_2 on the drawing, being located at the inlet and outlet of the cooler. The output of the cooler can then be obtained from the chart by reading the water flow pen and the two temperature pens and computing equivalent Btu. A second flow meter *2* is provided to measure the cold water output of the tank, and this meter also has two temperature pens; t_4 measures the warm return water from the system to

A method of improving air conditioning power load factors is to provide a means of storing refrigerating effect at night when the normal building electrical load is low and using the stored refrigeration to supplement compressor capacity during the day. Types of storage systems and operating results in a 19 story building were reported in the July number; this month, a description of a storage system in another building is given, with a discussion of how it is operated. . . . This information has been taken from a paper presented at the University of Texas air conditioning conference

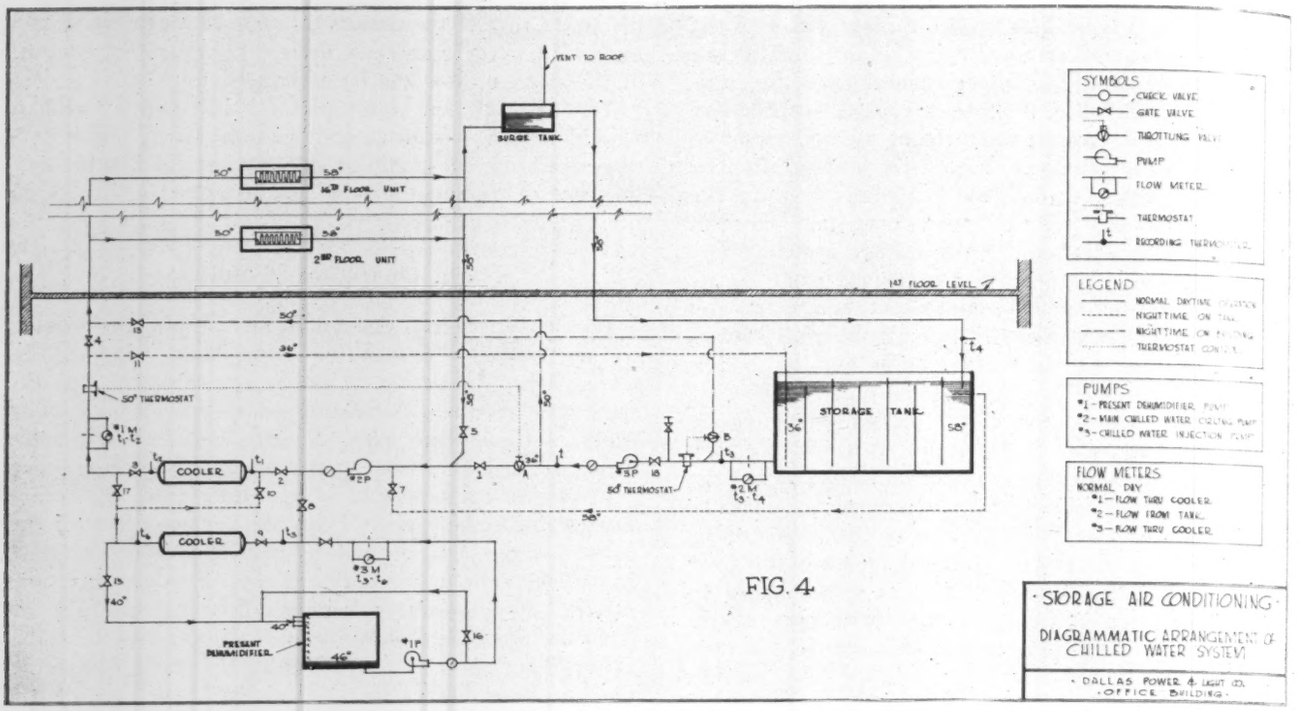


FIG. 4

the hot end of the tank and t_3 the cold water temperature leaving the tank. The Btu output of the tank can then be directly measured. A third flow meter designated as *3* on the drawing and having temperature pens t_5 and t_6 is provided on the older conditioning system. During the normal day operation this flow meter measures the Btu output of the centrifugal machine, which has no bearing on the storage system except when used at night to assist in cooling the tank.

A 24 point indicating thermometer is provided with its temperature elements located on each floor. By turning the dial on this instrument the operator in the basement can read the temperature at any point of the building. For the 10th floor it is possible to read both the wet and dry bulb temperatures, from which a representative inside condition is determined. It is also possible to read the outdoor wet and dry bulb temperatures. A 12 point balancing type thermometer calibrated to read within $\frac{1}{10}$ of 1 deg is supplied for determining the temperature of the water in each compartment of the storage tank.

Each of the fans in the equipment rooms throughout the building can be controlled from the basement by means of a remote control pushbutton station equipped with indicating lights. These fans can also be controlled from each equipment room.

Operation of Controls

The operation of the storage system is determined by the thermostatically controlled valve *A* which throttles the discharge from the injection pump *3* so as to allow a sufficient amount of 36 F stored water to be mixed with the 58 F return water, to give a 54 F temperature at the inlet to the chilled water cooler served by the ammonia compressors. Sufficient refrigerating capacity is provided to reduce the temperature to 50 F leaving the cooler under normal design conditions. If this water temperature exceeds 50 F, the action of the thermostat opens valve *A* to allow more 36 F water to enter the system. If the temperature to the units is below 50 F (or any predetermined setting) the thermostat closes valve *A* so as to allow less of the 36 F water to enter the system. As the injection pump forces 36 F water into the system, an equal amount of water is displaced in the surge tank, and an overflow line carries this water back to the warm end of the tank.

The storage tank is provided with temperature indicators at each of the passes between successive vertical baffles, and the operator determines the necessary capacity of refrigerating equipment by the relation of the temperatures in each pass to the time of the day. The system should be operated so as to take full advantage of the tank during the day and if, for example, at noon time the temperature readings were such as to indicate that there were but two of the seven passes available to supply 36 F water, it would show that the tank had been drawn upon too heavily during the forenoon and additional machine capacity must be put into service.

The thermostatically controlled valve *A* automatically proportions the mixture of cold and returned water, but the operator must use his judgment as to the amount of refrigerating capacity in service so that there will be a little reserve held in the storage tank at all times. Experience alone determines the best method of operation. The two ammonia compressors are each capable of 50 per cent reduction in capacity; consequently, four steps of refrigerating capacity are available at the operator's discretion.

A daily log sheet containing places for over 2000 entries of data is being kept. Some important questions, such as the daily and seasonal efficiencies of the storage tank, which are still viewed theoretically, will be answered this summer. By the end of the summer there will be available a very complete record for analysis of the operation of the semi-storage system.

How to Select and Install Spring Hangers for Piping

By W. F. Fischer*

HANGERS for the support of horizontal pipes of high pressure piping systems should not only be adjustable to facilitate the lining up of the piping system, but should also provide a means for taking up or relieving strains caused by expansion, contraction and vibration in other members of the system. The horizontal steam main shown in Fig. 1, for example,

The author has had many occasions in his engineering work to design spring hangers for piping of the kind discussed here. Because he could not find much published data on the subject, he prepared these tables and sketches, including in the tables a variety of spring sizes

is subjected to a vertical movement caused by the expansion and contraction of the vertical riser X, which is fixed at its upper end and expands downward 0.8 in. when heated, thus putting an additional strain on hangers *1* and *2*, and an upward thrust on the riser itself.

If the hanger springs are properly adjusted to bear the weight of the pipe, fittings and valves when cold, the expansion of the riser when heated will decrease the set height of each spring by 0.8 in., the amount of the expansion. If, therefore, the upward thrust on the riser shall not exceed the 800 lb specified, each of the two springs must be of such capacity that they will deflect (compress or shorten) not less than 0.8 in. under a load of 400 lb, or not less than 1 in. under a load of $400 \div 0.8 = 500$ lb. That is, the scale value V of the spring must not be greater than 500. (The **scale value** of a helical spring is that force in pounds required to compress or shorten the spring 1 in.)

Springs having a scale value of, say, 600 lb per in. deflection instead of 500 would shorten 1 in. under a load of 600 lb, or 0.8 in. under a load of $0.8 \times 600 = 480$ lb. In the case of Fig. 1, the reaction of the two springs would produce an upward thrust of $2 \times 480 = 960$ lb on riser X.

The **number of working coils** N in a helical spring is governed by the scale value V of the spring itself. Decreasing the scale value of the spring increases the number of working coils required, and also the free height of the spring H. Conversely, increasing the scale value of the spring decreases the number of working coils required and also the free height of the spring.

The **free height** of a helical compression spring is the overall length, or height, of the spring as wound before any load is applied to it. The **set height,** under any specific working load, is the overall length or height of the spring when bearing that load. The **solid height** h is the overall length or height of the spring when all coils are closed—metal to metal—under the maximum load for which the spring is designed. Springs should never be loaded to maximum capacity, except for test purposes, and when the test load is removed the spring should return again to its original free height.

The total amount of **compression**—or free space—in a helical compression spring is the difference in inches between the free height H and the solid height h measured parallel to the axis of the spring. The value of C depends on the number of working coils N and the pitch, or center to center distance of the coils, for its value (see Fig. 3). The **pitch,** or center to center distance of the coils, is equal to the diameter d of the wire or bar from which the spring is wound, plus the maximum allowable deflection F per coil. Helical springs are wound with one inactive coil at each end, but due to the tapering off of these two coils, they are counted as 1½ times the diameter of the bar d when measuring the total height of the spring parallel to its axis.

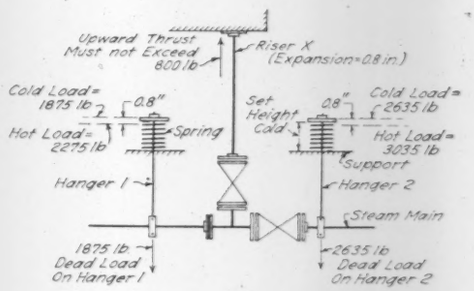


Fig. 1—Sketch illustrating example

Helical springs of best proportions are usually wound with a pitch diameter M equal to from four to seven times the diameter d of the bar. In no case should M be less than $3d$.

The capacity P, in pounds, and deflection F, per working coil, can be found from the following formulas:

$$P = \frac{\pi S d^3}{8M} \quad [1]$$

$$F = \frac{\pi S M^2}{Gd} \quad [2]$$

where P is the capacity of the spring, lb;
S is the fiber stress, at the most strained fiber of the bar, lb per sq in.;
d is the diameter of the bar, in.;
M is the mean, or pitch diameter of the helix, in. (found by subtracting the diameter of the bar from the outside diameter of the spring);
F is the deflection (or compression) of one working coil of the spring, in.;
G is the effective torsional modulus of elasticity of the material;
N is the number of active or working coils in the spring.

The total deflection (or compression) of the spring is equal to NF.

*The author has been connected with several of the large engineering concerns during the past 25 years.

Table 1—Maximum Capacity in Pounds (upper figures) and Deflection per Coil in Inches (lower figures) for Helical Springs of Round Steel Bars from ¼ In. to 11/16 In. in Diameter, Fiber Stress 100,000 Lb per Sq In.

Mean diam of Spring	Diameter of Bar					
	¼	5/16	⅜	7/16	½	9/16
1	613 0.120	1198 0.096				
1 1/16	577 0.135	1130 0.108				
1⅛	595 0.151	1068 0.121	1840 0.102			
1 3/16	517 0.169	1010 0.135	1743 0.113			
1¼	490 0.188	957 0.150	1655 0.125			
1 5/16	467 0.207	913 0.165	1575 0.138	2500 0.118		
1⅜	446 0.226	872 0.181	1505 0.151	2390 0.129		
1 7/16	427 0.248	835 0.198	1440 0.166	2290 0.142		
1½	408 0.270	800 0.215	1380 0.180	2195 0.154	3275 0.135	
1 9/16	392 0.293	767 0.234	1323 0.195	2105 0.167	3140 0.147	
1⅝	377 0.317	737 0.254	1273 0.212	2023 0.182	3024 0.159	
1 11/16	364 0.342	712 0.273	1228 0.228	1950 0.195	2910 0.171	4140 0.152
1¾	350 0.368	685 0.294	1182 0.246	1880 0.210	2810 0.184	4000 0.163
1 13/16	338 0.394	662 0.315	1130 0.263	1815 0.227	2705 0.197	3855 0.175
1⅞	327 0.421	640 0.337	1104 0.281	1755 0.241	2620 0.211	3680 0.187
1 15/16	316 0.450	620 0.359	1070 0.300	1700 0.257	2535 0.227	3606 0.200
2	306 0.480	598 0.383	1035 0.320	1648 0.274	2460 0.240	3500 0.214
2 1/16	297 0.512	581 0.408	1000 0.341	1593 0.292	2380 0.256	3390 0.228
2⅛	288 0.542	564 0.443	973 0.361	1547 0.309	2310 0.271	3285 0.241
2 3/16	280 0.574	550 0.458	946 0.383	1500 0.328	2240 0.287	3194 0.254
2¼	272 0.608	533 0.485	922 0.405	1464 0.347	2185 0.304	3105 0.270
2 5/16	264 0.641	515 0.512	890 0.428	1418 0.367	2150 0.321	3000 0.285
2⅜	258 0.677	505 0.540	873 0.452	1387 0.386	2068 0.339	2945 0.301
2 7/16	252 0.712	492 0.568	850 0.475	1352 0.406	2020 0.356	2875 0.316

Mean diam of Spring	Diameter of Bar							
	¼	5/16	⅜	7/16	½	9/16	⅝	11/16
2½	246 0.749	480 0.598	823 0.500	1317 0.428	1965 0.374	2800 0.333	3835 0.300	5110 0.272
2 5/8	234 0.827	456 0.660	788 0.551	1253 0.472	1870 0.413	2660 0.367	3650 0.330	4860 0.300
2¾	223 0.905	435 0.725	753 0.605	1192 0.517	1785 0.452	2545 0.403	3485 0.363	4635 0.330
2⅞	213 0.992	418 0.791	722 0.661	1148 0.566	1710 0.496	2435 0.440	3335 0.396	4445 0.360
3	204 1.080	400 0.862	690 0.720	1092 0.617	1637 0.540	2330 0.480	3195 0.432	4250 0.392
3⅛	196 1.170	383 0.934	662 0.781	1050 0.668	1568 0.585	2235 0.520	3065 0.467	4070 0.426
3¼	189 1.268	370 1.012	638 0.845	1012 0.723	1510 0.632	2152 0.562	2950 0.507	3925 0.460
3⅜		356 1.091	614 0.912	975 0.780	1453 0.683	2070 0.607	2840 0.545	3780 0.497
3½		343 1.173	591 0.980	940 0.838	1402 0.733	2000 0.653	2740 0.588	3645 0.534
3⅝		331 1.260	571 1.051	908 0.900	1355 0.788	1930 0.700	2644 0.629	3520 0.573
3¾			552 1.125	878 0.962	1310 0.843	1863 0.748	2555 0.674	3400 0.612
3⅞			533 1.203	848 1.030	1267 0.901	1798 0.800	2474 0.718	3290 0.655
4			518 1.280	823 1.092	1228 0.960	1748 0.852	2393 0.768	3185 0.697
4⅛				798 1.167	1190 1.022	1693 0.908	2324 0.815	3092 0.743
4¼				774 1.238	1156 1.085	1645 0.963	2255 0.867	3000 0.788
4⅜					1120 1.150	1596 1.040	2191 0.916	2915 0.835
4½					1090 1.217	1553 1.080	2130 0.973	2830 0.884
4⅝						1510 1.140	2070 1.025	2760 0.943
4¾						1472 1.200	2020 1.080	2693 0.982
4⅞						1432 1.267	1963 1.141	2620 1.036
5							1917 1.200	2560 1.091
5⅛							1870 1.262	2485 1.148
5¼								2430 1.200
5⅜								2375 1.261

Bar ⅝

Mean diam of Spring	Diameter of Bar		
	¾	13/16	⅞
2	4790 0.192		
2 1/16	4645 0.205	6187 0.185	
2⅛	4510 0.217	6000 0.197	
2 3/16	4380 0.230	5830 0.208	
2¼	4265 0.243	5680 0.221	
2 5/16	4120 0.257	5515 0.233	
2⅜	4040 0.271	5375 0.246	
2 7/16	3940 0.283	5235 0.259	

Bar ⅛

Mean diam of Spring	Diameter of Bar		
	¾	13/16	⅞
2	4790 0.192		
2 1/16	4645 0.205	6187 0.185	
2⅛	4510 0.217	6000 0.197	
2 3/16	4380 0.230	5830 0.208	
2¼	4265 0.243	5680 0.221	
2 5/16	4120 0.257	5515 0.233	
2⅜	4040 0.271	5375 0.246	
2 7/16	3940 0.283	5235 0.259	

Tables for Spring Calculations

Taking S equal to 100,000 lb per sq in., and G equal to 10,500,000 lb per sq in. as a maximum for a good grade of spring steel, the writer has computed the values of P and F as given in Tables 1 and 2. The values contained between the heavy stepped lines of the tables are those for springs of best proportions where the mean or pitch diameter of the spring varies from $4d$ to $7d$.

The upper figures in the body of the tables are those giving the value of P, the capacity of the spring in pounds for a maximum fiber stress S of 100,000 lb per sq in.; the lower figures give the deflection F per working coil under the maximum load P.

A helical compression spring, wound from a ⅞ in. diameter steel bar and having a mean diameter of 5¼ in. has a maximum capacity P of 5000 lb at 100,000 lb fiber stress, and will deflect 0.943 in. per working coil under that load (see Table 2). When winding this spring, the coils would be spaced or pitched ⅞ + 0.943 = 1.818, say 1.82 in., center to center, so that the load of 5000 lb when applied to the spring would compress it solid, bringing all coils in contact with one another, metal to metal.

Where there is but little vibration to contend with, a factor of safety of 1½ may be used when selecting springs from the tables; simply multiply the dead or working load to be applied to the spring by 1.5, and select a spring of that capacity from the table.

The actual deflection per coil under the working load which the spring is to carry in service can be found by dividing the deflection per coil given in the table by the factor of safety chosen. The number of working coils N can then be found by dividing the free movement required, for any given case, by the deflection per coil.

Selecting a Spring for a Dead Load Only

If in Fig. 1 there were no expansion to be cared for in designing the spring for hanger 2, we could assume a factor of safety of 1.5 and a free movement Q of, say, 3 in. under the dead load of 2635 lb. This would give a maximum capacity of $P = 1.5 \times 2635 = 3953$ lb at 100,000 lb per sq in. fiber stress.

From Tables 1 and 2 we find that we have a choice of five springs to select from—⅝x2 7/16, 11/16x3¼, ¾x4⅛, 13/16x5¼, or ⅞x6 in. Any one of these five springs will suit the purpose and will deflect about the same amount under the dead load. The ¾ in. x 4⅛ in. spring is perhaps the best choice. Its actual deflection per coil is $f = DF/P = (2635 \times 0.672)/4020 = 0.440$, number of working coils is $N = Q/f = 3/0.440 = 7$, total compression C under load P is $NF = 4.70$, and total compression under actual dead load D is $Nf = 3.08$. This spring has a scale value $V = P/C = 4020/4.70 = 856$; that is, it will deflect or shorten 1 in. under a load of 856 lb, or 3.08 in. under the dead load of 2635 lb for which it is intended. The total compression of 4.70 in. under maximum load P equals 4020 lb and brings all coils into contact, metal to metal.

Selecting Spring for Expansion or Contraction

The ¾ in. x 4⅛ in. spring as chosen above is quite suitable for hanger 2 of Fig. 1, provided there is no expansion to be cared for in riser X. If, however, the same spring is used to care for the expansion as well, and we provide a spring having the same scale value ($V = 856$) for hanger 1, the reaction of the two springs when the riser expands 0.8 in. would produce an upward thrust of $2 \times 856 \times 0.8 = 1370$ lb on riser X, instead of the permissible thrust of 800 lb specified.

In order to use this same spring for the purpose intended we must increase the number of working coils N to give the spring a scale value of 500 or less. The number of coils required is, therefore $N = P/VF = 4020/(500 \times 0.672) = 12$ working coils. The total compression of the spring under the maximum load $P = 4020$ lb is then $C = NF = 12 \times 0.672 = 8.064$ in. and the true scale value of the spring is $V = 4020/8.064 = 498.51$. Thus, this ¾ in. x 4⅛ in. helical spring

†Q = free movement desired = approximately 3 in. (Chosen arbitrarily by the designer in order to find the number of working coils.)

having 12 working coils, is suitable for the conditions given.

The solid height h of this spring is $h = dN + 1.5d = d(N + 1.5) = 0.75 (12 + 1.5) = 10.125$ in. The free height of the spring is, therefore, $H = h + C = 10.125 + 8.064 = 18.19$ in.

The dead load D carried by the spring before the riser starts to expand is 2635 lb, and the set height due to this load is $A = H - (D/V) = 18.19 - (2635/498.51) = 12.91$ in.

The additional load w on the spring due to the expansion e of riser X is $w = Ve = 498.51 \times 0.8 = 398.81$ lb. The maximum working load W to be carried by the spring when the riser is hot is, therefore, $W = D + w = 2635 + 398.81 = 3033.81$ lb, and the set height due to this working load is $B = A - e = 12.91 - 0.8 = 12.11$ in.

The extreme fiber stress on the material of the spring due to the dead load D alone is given by $J = DS/P = (2635 \times 100{,}000)/4020 = 65{,}547$ lb per sq in. This gives a factor of safety of 1.53 approximately.

The extreme fiber stress due to the working load W is given by $K = WS/P = (3033.81 \times 100{,}000)/4020 = 75{,}468$ lb per sq in., giving a factor of safety of 1.33.

An extreme fiber stress under 80,000 lb per sq in. can be considered safe for a helical wound compression spring wound of a good grade of spring steel and supporting a steady, non-vibrating load.

Table 2—Maximum Capacity in Pounds (upper figures) and Deflection per Coil in Inches (lower figures) for Helical Springs of Round Steel Bars from ¾ In. to 1⅛ In. in Diameter, Fiber Stress 100,000 Lb per Sq In.

Mean diam of Spring	Diameter of Bar					
	¾	13/16	⅞	15/16	1	1 1/16
2¼	7365 0.203					
2⅜	6975 0.226					
2½	6625 0.249	8450 0.231				
2⅝	6300 0.276	8020 0.254	10022 0.235			
2¾	6025 0.302	7652 0.279	9570 0.259			
2⅞	5770 0.331	7340 0.305	9150 0.282	11255 0.264		
3	5520 0.359	7020 0.332	8765 0.309	10755 0.288	13100 0.269	
3⅛	5285 0.390	6730 0.360	8418 0.334	10320 0.312	12530 0.293	
3¼	5097 0.421	6480 0.390	8090 0.362	9950 0.338	12083 0.316	
3⅜	4905 0.455	6230 0.420	7795 0.389	9580 0.364	11630 0.342	13950 0.321
3½	4733 0.489	6025 0.452	7520 0.432	9240 0.392	11220 0.366	13500 0.345
3⅝	4570 0.526	5815 0.485	7257 0.449	8920 0.420	10825 0.394	13000 0.371
3¾	4418 0.561	5617 0.518	7020 0.482	8620 0.449	10472 0.420	12560 0.396
3⅞	4270 0.601	5430 0.554	6789 0.513	8330 0.481	10120 0.450	12150 0.424
4	4142 0.638	5270 0.591	6570 0.548	8080 0.512	9817 0.479	11780 0.451
4⅛	4020 0.672	5100 0.628	6377 0.582	7835 0.545	9530 0.511	11420 0.480
4¼	3898 0.720	4950 0.667	6180 0.620	7600 0.578	9240 0.540	11100 0.509
4⅜	3782 0.765	4815 0.706	6013 0.654	7400 0.612	8980 0.574	10760 0.539
4½	3681 0.808	4675 0.748	5840 0.695	7180 0.648	8726 0.606	10470 0.572
4⅝	3580 0.855	4550 0.789	5688 0.731	6985 0.685	8485 0.642	10200 0.603
4¾	3488 0.900	4435 0.832	5540 0.772	6800 0.721	8267 0.675	9930 0.636
4⅞	3395 0.950	4315 0.887	5396 0.812	6630 0.761	8050 0.712	9670 0.670
5	3313 0.997	4210 0.923	5265 0.857	6470 0.800	7854 0.748	9435 0.705
5⅛	3230 1.050	4100 0.970	5133 0.898	6300 0.835	7650 0.788	9180 0.740

Mean diam of Spring	Diameter of Bar						
	¾	13/16	⅞	15/16	1	1 1/16	1 ⅛
5¼	3155 1.099	4015 1.017	5000 0.943	6150 0.880	7480 0.824	8970 0.776	10650 0.735
5⅜	3080 1.157	3920 1.067	4894 0.987	6020 0.925	7300 0.867	8770 0.815	10402 0.768
5½	3005 1.210	3812 1.116	4770 1.038	5870 0.968	7140 0.905	8550 0.853	10160 0.808
5⅝	2940 1.270	3755 1.170	4670 1.085	5750 1.012	6980 0.950	8375 0.892	9930 0.843
5¾	3660 1.220	4575 1.135	5620 1.058	6835 0.993	8200 0.935	9730 0.883	
5⅞	3480 1.273	4470 1.182	5500 1.103	6675 1.033	8000 0.972	9500 0.920	
6	4385 1.237	6550 1.080	7850 1.015	9325 0.960			
6⅛	4290 1.287	5275 1.200	6415 1.127	7700 1.058	9120 1.020		
6¼	4210 1.340	5170 1.250	6280 1.171	7535 1.100	8950 1.043		
6⅜	4125 1.395	5070 1.302	6160 1.220	7400 1.147	8760 1.085		
6½	4965 1.351	6050 1.268	7250 1.190	8600 1.128			
6⅝	4880 1.404	5920 1.316	7115 1.238	8425 1.170			
6¾	4790 1.460	5815 1.370	6980 1.286	8275 1.217			
6⅞	5720 1.420	6850 1.332	8135 1.260				
7	5615 1.470	6740 1.380	7985 1.308				
7⅛	5500 1.520	6600 1.429	7840 1.352				
7¼	5420 1.580	6500 1.483	7710 1.405				
7⅜	5330 1.632	6400 1.533	7590 1.451				
7½	6280 1.587	7450 1.490					
7⅝	6175 1.640	7330 1.533					
7¾	6080 1.695	7200 1.605					
7⅞	5990 1.750	7100 1.658					
8	6985 1.710						
8⅛	6875 1.765						

Selection of Springs for Riser Hangers

Long vertical pipes and risers subject to expansion and contraction should be supported by flexible hangers or supports. If a long riser is supported at certain intervals by rigid hangers, properly adjusted so that each hanger will bear its share of the load when the riser is cold, the riser will expand upward when heated, taking the weight off the upper hangers or supports, in which case the lower hanger will have to bear practically all of the load. Such a case is illustrated in Fig. 2. The riser is 105 ft long and is supported at its lowest end on a base elbow, as shown, and at three other points by hangers marked 1, 2 and 3. The pipe, with its covering, weighs 90 lb per ft and the dead load on each hanger is $30 \times 90 = 2700$ lb. The expansion in 30 ft of pipe is 0.6 in.

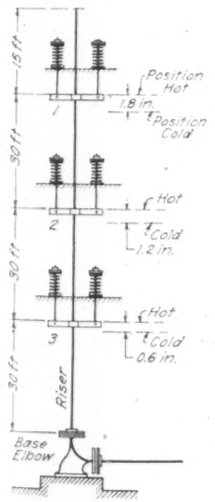


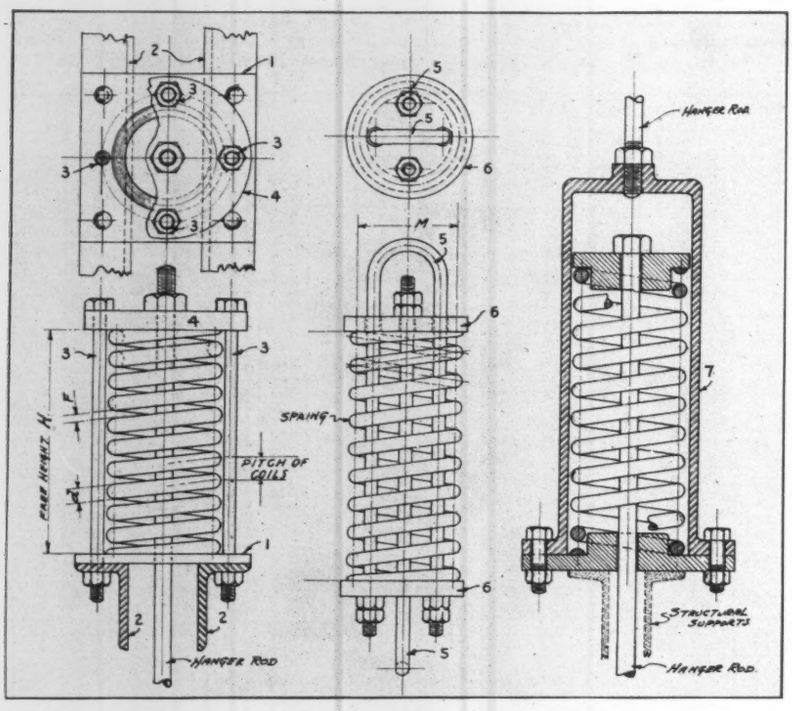
Fig. 2—Sketch for example on selection of riser hangers

Since two springs are to be used for each hanger, the dead load on each spring is 1350 lb. If all three hangers are properly adjusted for this share of the load when the riser is hot, the contraction of the riser when cold will cause an additional compression of 1.8 in. on each spring for hanger 1, 1.2 in. on those for hanger 2, and 0.6 in. on those for hanger 3.

Taking an arbitrary figure of, say, $1350 + 450 = 1800$ lb as the working load W on each spring with the riser cold, we see that each of the two springs for hanger 1 must compress or shorten 1.8 in. under a load of 450 lb, or 1 in. under a load of not less than $450/1.8 = 250$ lb, the scale value V for the springs for hanger 1. In like manner, we find the scale value V for the two springs for hanger 2 to be $450/1.2 = 375$ lb per in. deflection, and that for the two springs for hanger 3 to be $450/0.6 = 750$ lb per in. deflection.

Taking a factor of safety of 1½, and a maximum working load of 1800 lb, as above, a spring having a capacity of $P = 2700$ lb will be required.

Table 1 gives a choice of four springs, but of the four a 11/16 in. x 4⅝ in. spring, having a capacity of 2760 lb at 100,000 lb fiber stress, and a deflection of 0.943 in. per coil, seems the best suited for the purpose. Taking a trial value for $C = 2760/250$ for the springs for hanger 1, $C = 2760/375$ for those for hanger 2, and $C = 2760/750$ for those for hanger 3, and using the next



Figs. 3, 4 and 5 (left to right)—Methods of supporting and guiding springs

larger number of coils as given by $N = C/F = C/0.943$, we find that the springs for hanger *1* will require 12 working coils, those for hanger *2* eight working coils; and those for hanger *3*, four working coils.

The true value for C, then, will be $12 \times 0.943 = 11.32$ in. for the springs for hanger *1*, $8 \times 0.943 = 7.54$ in. for the springs for hanger *2*, and $4 \times 0.943 = 3.77$ in. for the springs for hanger *3*. These values are given in the eighth line of Table 3.

By adding the true values of C as found above to the solid heights h as given in the sixth line of Table 3, we get the free height of each spring as given in the seventh line.

The true scale value V for each spring is found by dividing P, the maximum capacity of the spring in pounds, by the true value of C for each spring. These values of V are given in the ninth line of Table 3.

If all springs are adjusted for a dead load of 1350 lb when the riser is hot, or to the set heights given in the eleventh line of the table, an additional compression of 1.8 in. in the springs for hanger *1*, 1.2 in. on those for hanger *2*, and 0.6 in. on those for hanger *3*, when the riser cools down, will increase the load on each spring by practically the same amount, or approximately by 430 lb, as given in the thirteenth line of the table. If, however, the final spring adjustments are to be made when the riser is cold, or when erecting the piping system, all springs should be set for a load of 1789 lb, or to the set heights as given in the fifteenth line of the table. If the adjustments are made as outlined, each spring should bear its proper share of the load at all times, whether the riser is hot or cold.

Table 3—Springs for Riser Hangers Shown in Fig. 2

Line No.		Hanger Marked		
		1	*2*	*3*
1	Diameter of bar, d, in.	11/16	11/16	11/16
2	Mean, or pitch diameter of spring, M, in.	4 5/8	4 5/8	4 5/8
3	Maximum capacity (at 100,000 lb fiber stress), P, lb.	2,760	2,760	2,760
4	Deflection F per coil under load P, in.	0.943	0.943	0.943
5	Number of working coils, $N = C/F$	12	8	4
6	Solid height of Spring, $h = d(N+1.5)$, in.	9.28	6.52	3.78
7	Free height of spring, $H = h + NF$, in.	20.60	14.06	7.55
8	Total compression under load P, $C = NF$, in.	11.32	7.54	3.77
9	Scale value of spring (lb per in. deflection), $V = P/C$	243.82	366.05	732.10
10	Dead load (no expansion or contraction), D, lb.	1,350	1,350	1,350
11	Set height due to dead load D, $A = H - D/V$, in.	15.10	10.37	5.71
12	Compression of spring caused by contraction of riser, e, in.	1.8	1.2	0.6
13	Added load on spring due to C, $w = Ve$, lb.	438.88	439.26	439.26
14	Working load (riser cold), $W = D + w$, lb.	1,789	1,789	1,789
15	Set height of spring due to working load, $B = A - e$, in.	13.30	9.17	5.11
16	Fiber stress due to dead load, $J = DS/P$	48,913	48,913	48,913
17	Fiber stress due to working load, $K = WS/P$	64,820	64,820	64,820

Adjusting, Supporting and Guiding Springs

Long helical compression springs should be guided in some suitable manner to prevent buckling or bending sideways when loaded. For springs of small mean or pitch diameter M this may be taken care of by the hanger rod which is placed on the inside of the spring. The inside diameter of a ½ in. x 2 in. mean diameter spring, for example, is only 1½ in., in which case a 1¼ in. diameter hanger rod can be used for the purpose. For springs of mean diameter larger than 2 in., some such scheme as that illustrated in Figs. 3, 4 and 5, may be employed to prevent buckling. The spring illustrated in Fig. 3 is supported on a flat plate *1* attached to structural members *2*. Four round rods *3* are fitted loosely in holes in the upper plate *4*, spaced uniformly around the outside of the spring to form a cage which prevents buckling when the spring is compressed.

An effective method of guiding the spring when it is to be suspended from some overhead member is that shown in Fig. 4. Two U-shaped rods marked *5* passing through holes in the upper and lower container plates *6* provide an effective guide for the spring, and at the same time a place to attach the hanger rods. Hanger rods with an eye at one end may be used for the purpose and the U rods slipped through them when assembling the hanger in the field. In Fig. 4, the U rods are shown inside the spring, but for springs of smaller mean diameter the rods can be placed on the outside.

A somewhat more elaborate method of accomplishing the desired result is that shown in Fig. 5. In this case the spring is contained inside a metal cylinder or cage *7* which may be made from a piece of standard steel pipe or boiler tubing with a flange welded on one end and the other end closed as shown.

Since the mean diameter of the spring tends to increase slightly when the spring is loaded, a clearance of about ⅛ in. should be allowed between the outside of the spring and the inside of the cage or cylinder.

Preparing Contract Documents for Low Rent Housing Projects

PREPARING the contract documents is the subject of the third part of the United States Housing Authority's bulletin on the heating of low rent housing projects:

All heating plans and specifications should be in sufficient detail so that a minimum of shop drawings will be necessary.

A. The Working Drawings

Ample space should be provided for piping and particular care be taken to avoid structural interferences and conflicts between the different branches of the mechanical trades. Mechanical drawings should be generally diagrammatic, with all piping and equipment shown as nearly as possible in the location in which it is to be installed.

The use of a common trench for buried heating and hot water lines wherever possible should be encouraged.

Pipes should not be located in exterior walls where there is danger of freezing, unless proper protection is provided.

Pipe sleeves and openings should be shown on the structural or architectural drawings for all pipes passing through footings, masonry floors or exterior walls below grade. The elevations of sleeves need not be given except where passing through waterproofing finishes, which require special sleeve construction. In full basements there should be provided not less than 6 ft 3 in. clear head room under bottom of lowest pipe.

B. The Specifications: Materials

Black steel pipe: all steam mains and all dry returns.

Wrought iron pipe: all wet returns, all underground returns, and boiler feed and boiler blow-off lines.

Copper molybdenum alloy pipe: all wet returns, all underground returns, and boiler feed and boiler blow-off lines.

Extra strong steel pipe: all wet and underground returns.

Cast fittings: threaded or flanged cast iron or brass of cast iron steam pattern.

Welding fittings: wrought iron or copper moybdenum alloy may be used interchangeably for wrought iron or copper molybdenum pipe. Welding elbows should have a face to center dimension equal to not less than 1½ nominal diameters of the pipe to which they are connected.

Pipe joints: in buildings—joints 1¼ in. and smaller, screwed; 1½ in. to 6 in. inclusive, screwed or welded; 8 in. and larger, flanged or welded; underground—1½ in. and larger, welded.

Valves: ¾ in. and smaller, globe valves; 1 in. and larger, generally gate valves; 2 in. and smaller, brass; 2½ in. and larger, iron body brass mounted; 5 in. and smaller, screwed; 6 in. and larger, flanged.

Brass metal for globe valves should have a minimum copper content of 80 per cent and a maximum lead content of 5½ per cent.

C. Outline of a Typical Specification

1) General note—Covering general conditions, and other sub-divisions of the specifications.
2) Correction of faulty materials and workmanship.
3) Work included, schedule of general requirements; work not included.
4) Drawings—schedule, correlation to specifications, and shop drawings.
5) Testing requirements, prior tests, inspections, guarantee.
6) Temporary heat requirements.
7) Removal of debris during and at completion of contract.
8) Coördination of work with other trades.
9) Permits, certificates of approval, etc.
10) Excavation and backfill, grades and elevations.
11) Description of installation and materials for heating system within building.
12) Description of installation and materials for heating system outside of building.
13) Description and installation of boilers; type, construction, capacity, meters, and controls.
14) Foundations, pits for boilers, and other concrete work for equipment.
15) Smoke breeching—dampers, supports.
16) Boiler trimmings and tools.
17) Refractory, sand, cement, lime, etc.
18) Chimney construction and draft available.
19) Oil burners.
20) Oil pumps—motors, controls, and description of piping.
21) Oil heaters, relief valves, oil tanks, meters.
22) Soot blowers, type and description.
23) Safety devices for boilers, oil burners, and water supply.
24) Exhaust heads.
25) Vacuum pumps, motors, controls.
26) Pressure reducing valves, strainers, safety valves.
27) System of heating controls.

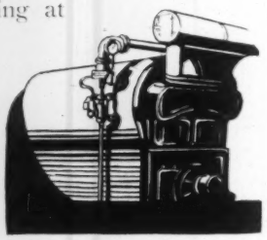
The third part of the United States Housing Authority's bulletin on the heating of low rent housing projects given here covers preparation of the contract document. . . . The introductory paragraphs of the bulletin, and Part 1 on economic analysis, were published in the June HPAC. Part 2, giving recommendations for design of heating plants, was published in July

28) Radiation.
29) Radiator valves, thermostatic traps, radiator hangers.
30) Heating supply mains.
31) Return mains—steam traps, combination float and thermostatic traps.
32) Pipe and fittings.
33) Joints.
34) Unions, gaskets, bolts, and nuts.
35) Valves, escutcheons.
36) Metal U covers for pipes in fill.
37) Pipe nipples.
38) Open ends.
39) Pipe sleeves.
40) Trenches and covers.
41) Hangers, anchors—guides, inserts, etc.
42) Steam connections to hot water generators.
43) Expansion provisions.
44) Welding for pipe and fittings.
45) Insulation for boilers and other equipment.
46) Insulation for boiler room, including hanging ceiling if necessary, and ventilating equipment.
47) Painting.
48) Unit heaters, motors, controls.
49) Ventilating fans, motors, controls.
50) Exhaust fans, motors, controls.
51) Sheet metal work.
52) Grilles, registers.
53) Factory inspections.
54) Testing boilers, pumps, fans, etc.
55) Testing of piping within building.
56) Testing of underground distributions.
57) Special equipment, accessories.
58) Voltage and other characteristics of electrical service.

D. Steam Space Heaters

Include all necessary steam and return connections, thermostatic traps and shut-off valve, and an automatic controlled valve.

The heating element (of non-ferrous tubes and fins, with ferrous or non-ferrous headers) should be suitable for a steam working pressure of 2 lb per sq in. gage with air entering at 67 F. When operating at full speed and maximum capacity, the final temperature should not exceed 125 deg; when operating at less than full speed and maximum capacities specified, the final temperature should be not less than 110 F.



With the fan off, the space heater with the thermal head available should be capable of releasing approximately 25 per cent of the total heat output. It should be of the blow through type with direct connected motor and fan assembly, consisting of one or more forward curved double inlet fans mounted on the extended shaft of a three speed motor. The motor must be quiet running regardless of the speed at which it operates.

Use standard air, measured at the inlet, for the cubic feet of air per minute to be recirculated through the space heaters.

Complete suggested specifications may be obtained upon application to the United States Housing Authority, Washington, D. C.

E. System of Heating Controls

The purpose of the heating control system is to provide a heating system in which the circulation of steam will be equalized either by an adjustable orifice made as an integral part of the radiator valve or as a properly sized removable regulating plate in each radiator valve assembly. Any other means may be used which will insure receipt of steam by all radiators at the same time, proportionately. The systems of zone control may be as follows:

1) *Continuous Flow System*

a) Floating or throttling type.

b) A valve which functions in progressive steps of opening and closing.

Description of Continuous Flow System

a) To be capable of circulating steam at variable sub-atmospheric steam pressures and steam temperatures by means of a floating or throttling type continuous flow electric or air operated control valve taking steam direct from supply at up to 15 lb pressure above atmosphere.

b) To be capable of controlling the quantity of steam supplied so as to heat the radiators fractionally to vary their heat output by means of a continuous flow electric or air operated control valve taking steam direct from supply at up to 15 lb pressure above atmosphere.

Panel Boards

1) To be of the automatic type with heat balancer and selector or other suitable equipment.

2) To be of the automatic type with outside thermostat and selector.

2) *On and Off System*

a) A valve that is opened and closed periodically.

b) A valve that functions to produce pulsating steam flow.

Description of On and Off System

a) To be capable of controlling automatically the quantity of steam supplied by intermittently opening and closing the control valve for fixed cycles during which the valve is opened automatically selected by an outside thermostat, electric or air operated control valve taking steam from supply at up to 15 lb pressure above atmosphere.

b) To be capable of controlling automatically the quantity of steam supplied by intermittently opening and closing the control valve for periods of time with these periods automatically selected by a combination of outside and radiator temperatures or outside and inside temperatures, electric or air operated control valve taking steam from supply at up to 15 lb above atmosphere.

Panel Boards for both (a) and (b) to be of the automatic type to synchronize the action of the several zones.

Manual Control should be provided to increase or decrease the heat flow variably, for producing continuous steam flow and for shutting off the steam flow entirely and provide means for automatically shutting off the steam flow when the outside temperature rises above 65 F.

3) *The Combination System* will include the principal functions of both No. 1 and No. 2 described heretofore.

Requirements for All Systems

Each of the above systems should be balanced to synchronize the action of the several zones so as to produce an even load curve on the boiler plant. Demonstration

[*Concluded on page 506*]

Heat Transfer and Viscosity Effects in Ammonia Shell and Tube Brine Coolers

By Richard J. Panlener*

THIS review of data and tests on heat transfer in horizontal shell and tube coolers used for cooling calcium chloride brine is continued from Part 1, which was published in the June HPAC.

Boiling Point Depression

The reduced evaporating temperature due to the hydrostatic head of the refrigerant in the flooded brine cooler has been long recognized as a problem in flooded coolers. Although it is not strictly accurate, it is a satisfactory approximation, for purposes of calculating heat transfer, to assume the average temperature of the boiling liquid to be that at a point midway between the top and the bottom of the boiling mass.

The very fact that this head presents variable evaporating pressures and temperatures in the cooler may cause a variation in the ammonia film, gas formation, and convection currents. However, the major effect of this refrigerant hydrostatic head is upon the evaporating temperature and hence upon the ultimate cooler capacity.

Average Brine Viscosity

Fluid flow has been resolved into two clear and distinct types, streamline and turbulent. There is a zone between these two which is the critical region and in which either streamline or turbulent flow may occur depending upon how it is approached. Only turbulent flow will be considered, since the velocities necessary and heat transfer coefficients obtained under streamline flow would be impractical for commercial applications.

The type of flow is readily determined by the use of Reynolds number, which is a function of the inside diameter of the tube, the velocity, the density and the absolute viscosity of the fluid at the temperature and pressure conditions under which the flow occurs. Since the viscosity values for calcium chloride were available over a wide range of brine concentrations and temperatures, a series of tests was run to ascertain the feasibility of determining the pressure drops through horizontal multipass coolers with this method. The major question to settle was the effect of the brine heads. For this reason, a 10 pass cooler was used for the tests so that any discrepancy brought about by the heads would be magnified. The velocity range in the tests extended from values below to values far above those encountered in commercial applications. Measurements of brine temperatures and quantities, and pressures at inlet and outlet, were carefully made. Fig. 5 shows the curves for both the test data and calculated values derived by means of Fanning's formula. The test values represent accurate readings obtained both by calibrated gages and by mercury manometer. The calculated values which superimpose the test values were obtained by using the friction factor for drawn steel tubing in Fanning's formula. The upper curve of calculated values represents the pressure drop values obtained when the friction factor for clean steel or cast iron pipe was used in Fanning's formula. This curve indicates values which are approximately 12 per cent higher, and more accurately approaches the true operating pressure drop through a cooler after incrustation and tuberculations have formed on the inside of the tubes. It would also allow for variations which occur in design problems of the brine cooler heads where fluid flow restrictions are not infrequently encountered.

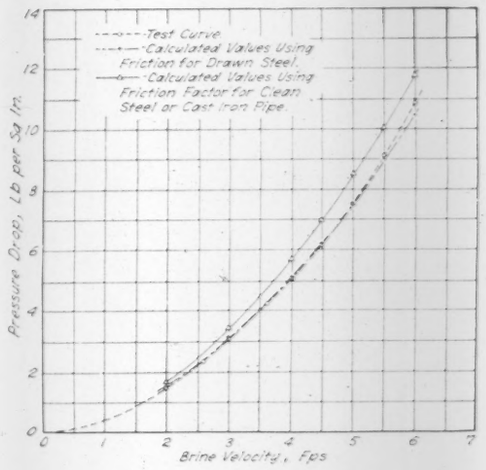


Fig. 5—Brine friction as determined by test and as calculated by means of Fanning's formula

These tests also showed that a cooler operating with a constant brine velocity, a constant MTD, but a variable brine temperature has a 50 per cent increase in pressure drop as the temperature of the brine off the cooler

A thorough analysis of the factors influencing the overall heat transfer and friction values in horizontal multipass flooded ammonia brine coolers is presented by the author. After discussing the effects of the individual variables, an empirical heat transfer formula and accompanying curve give a simple and rapid method of selecting the heat transfer coefficients. A description of the determination of friction values by Reynolds numbers and an accompanying test curve to further establish its accuracy are also included. . . . The data are of interest to engineers concerned with refrigeration piping, and in connection with brine spray equipment for low temperature air conditioning

*Vilter Mfg. Co.

Part 2. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, June, 1939, pp. 351-353.

changes from 40 to —30 F. Since the diameter of the tube and the brine velocity remained constant, it is obvious that the total effect was brought about by the changes in the brine viscosity and brine density.

Having determined the viscosity effect upon friction, the tests were analyzed to show how the heat transfer varied with the average viscosity. Fig. 6 indicates this change in heat transfer with a constant brine viscosity and MTD and a variable brine velocity.

Kratz, Macintire, and Gould (*7*)† have approached the problem of the effect of the water film upon the heat transfer coefficients in horizontal ammonia multipass condensers. The problem is attacked with the idea that heat is transmitted from a flowing fluid to the metallic surface through the stationary fluid film. The thickness of this film and hence the conduction through it depends upon the viscosity of the fluid.

Although test information does not permit the individual coefficients to be broken down and resolved into their individual values, Fig. 7 shows the great effect of

†See bibliography.

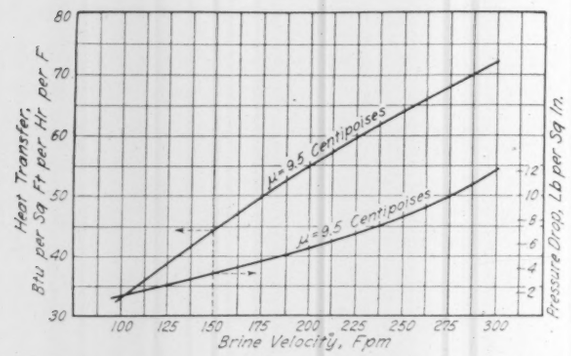


Fig. 6—Effect of a variable velocity on the heat transfer and pressure drop when the viscosity and MTD remain constant

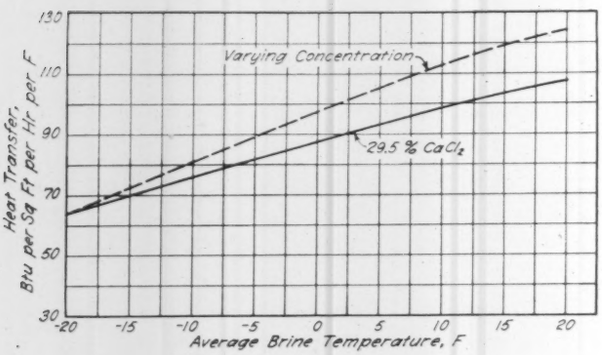


Fig. 8—Effect of brine concentration

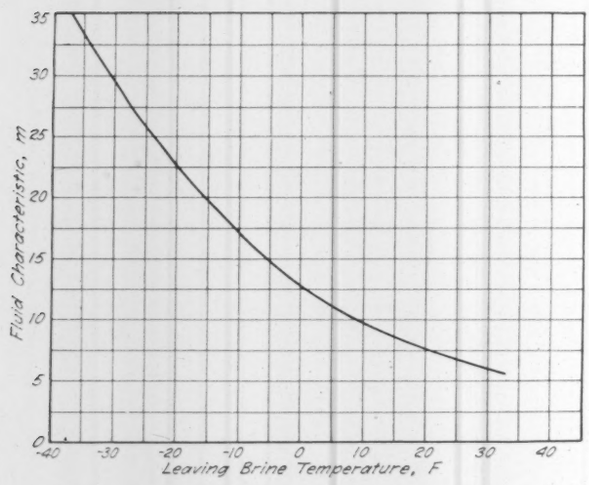


Fig. 10—Values of fluid characteristic *m* plotted against leaving brine temperature

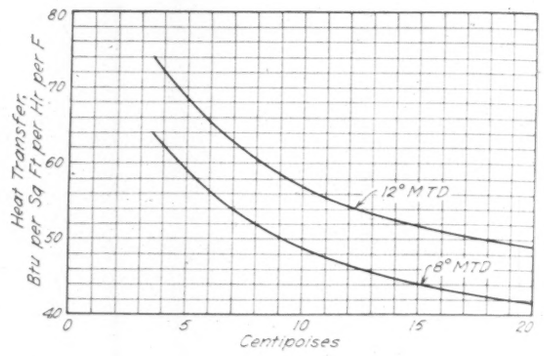


Fig. 7—Variation of heat transfer coefficient with changes in MTD and average brine viscosity

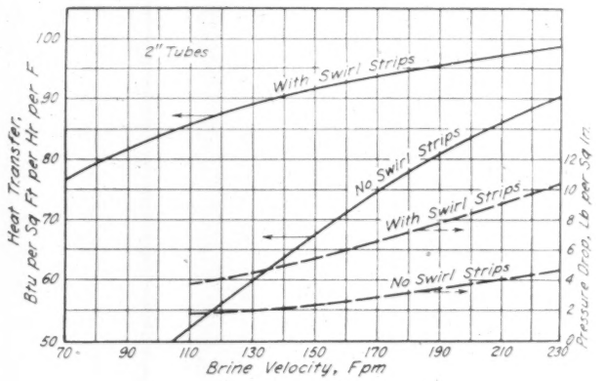

2" Tubes
With Swirl Strips
No Swirl Strips
With Swirl Strips
No Swirl Strips
Heat Transfer, Btu per Sq Ft per Hr per F
Pressure Drop, Lb per Sq In.
Brine Velocity, Fpm

Fig. 9—Effect of swirl strips on heat transfer at various velocities

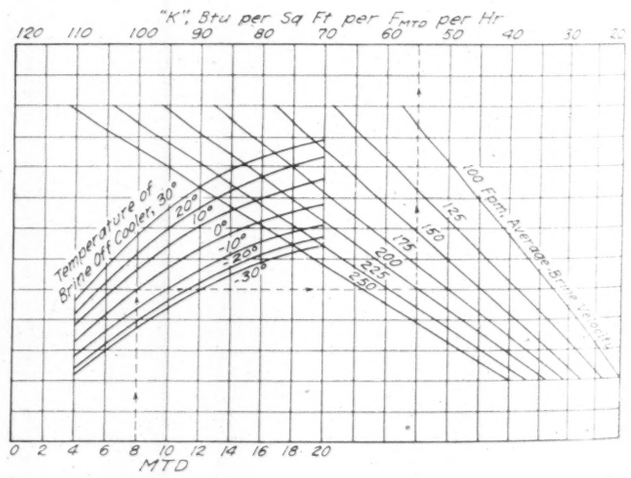


Fig. 11—Heat transfer curve for flooded ammonia horizontal multipass shell and tube brine coolers with 2 in. OD seamless steel tubes

viscosity. As the viscosity values increase, the brine film agitation lessens and the fluid flow becomes less turbulent. Also, there is undoubtedly less activity in the buffer layer since the increased viscosity would decrease the eddy mixing. Fig. 6 graphically shows the combined variation in the heat transfer coefficient and the fluid pressure drop through the cooler when the viscosity and MTD are constant and the velocity changes. The similarity of the curves definitely indicates a relationship between the two and the need for viscosity consideration in heat transfer values.

Consley's report (*2*) covers a series of tests which were performed under varying conditions of velocity and temperature but with only one concentration, namely, 1.28 specific gravity brine at 60 F. He worked "on the assumption that the gain in heat transfer does not warrant the risk of a freeze-up in a commercially operated plant, particularly where lower brine temperatures are encountered." However, he includes Fig. 8 to illustrate the possible increase in heat transfer with correct concentrations. Fig. 6 also shows the effect of the heat transfer as brought about by a variable velocity and a constant viscosity, or in other words, indicates the variation in heat transfer with a change in the brine film. Consley does not include a correction curve for the heat transfer when the proper brine concentration is used and the velocity varied.

Effect of Swirl Strips

The effect of swirl strips on heat transfer values has long been disputed in commercial application work. Consley shows in Fig. 9 with varying "apparent" velocities the effect of swirl strips both upon the heat transfer values and upon the brine pressure drop through the cooler. Again all these test values are for 1.28 specific gravity brine at 60 F and for 9.5 deg MTD.

To understand the effect of swirl strips, the brine flow must be analyzed. When swirl strips are placed inside the tube the brine travels in a spiral path which results in greater agitation or turbulence as well as higher velocities. These higher velocities again reduce the brine film resistance, the extent of which depends upon the quantity of brine circulated and the pitch of the swirl. However, these higher velocities result in higher friction values, all of which is not beneficial. The additional friction is partly made up by the additional surface of the swirl strip, which of course has its brine film and accompanying resistance to flow.

When a comparison is made between two identical coolers with the same size tubes but into one of which swirl strips have been placed, the true conclusion cannot be forthcoming. To analyze properly the value of this accessory equipment, tests should be run on two similar coolers under identical conditions of specific gravity, viscosity, mean temperature difference, and "actual" brine velocity. To compare values with the 2 in. OD tubes as shown, it would be necessary to use another dynamically similar cooler with smaller tubes so that with the same quantities of brine, which would result in higher velocities, identical brine film conditions would be obtained.

Additional factors would enter into this attack on the problem. First there would be an increase in the heat transfer factor due to the smaller tube, as was previously discussed. Second, the additional increase in friction would be brought about entirely by the brine film on the tube wall only; there would be no friction loss attributable to any other resistance to the fluid flow. The problem of comparison of these two types of coolers is a difficult one and on the basis as outlined it may be possible to reorganize existing test data to obtain a conclusive answer.

Conclusion

After thoroughly investigating available heat transfer data for flooded ammonia horizontal multipass shell and tube brine coolers (without swirl strips) it was possible to obtain the following empirical formula:

$$K = 2.97 \frac{V^{0.700}}{D^{0.4} \times m^{0.408}} \times \mathrm{Log}_{10}\,(d+2)$$

where

K = Btu per sq ft per F MTD per hr (overall heat transfer);
D = Outside diameter of tubes, in.;
V = average brine lineal velocity, fpm;
d = logarithmic mean temperature difference;
m = fluid characteristic depending on specific gravity, internal thermal conductivity, and viscosity. Values for m plotted against leaving brine temperature are given in Fig. 10.

The plot for the above formula for 2 in. OD tubes is shown in Fig. 11, and presents a less complicated method for obtaining the overall heat transfer value. For example, if a cooler were to deliver 0 F brine with a mean temperature difference of 8 deg and an average brine velocity of 200 fpm, the heat transfer would be obtained by proceeding vertically from the 8 deg on the MTD scale to the line for 0 F brine temperature off the cooler, then horizontally to the 200 fpm average brine velocity line, then again vertically to the reversed heat transfer scale where 55 Btu per sq ft per F MTD per hr is read.

When one considers the ramifications of the problems represented by the flooded ammonia horizontal multipass shell and tube brine cooler, it is easily understood why overall heat transfer coefficients are used. Only recently have efforts been made to apply dimensional analysis to these types of refrigeration problems and it was not until this method was applied that the multitude of variables was subjected to determination by testing. However, there still remains additional fundamental research to be obtained and testing and coördination to be done.

[*Concluded*]

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Current Types of Gaskets and Closures

By **Parameter**

A STORY has been told concerning a Spanish peasant who had been hired to transport half a dozen small packages, weighing perhaps 30 lb apiece. Thereupon he put all of them into one of the panniers upon his donkey's back and presently noticed that the panniers had slipped and the load had changed position. After deep thought, he filled the empty pannier with stones, levelled the loads, mounted his beast and rode away contentedly.

And it came to pass that a descendant of this man became an engineer who had been hired to design a high

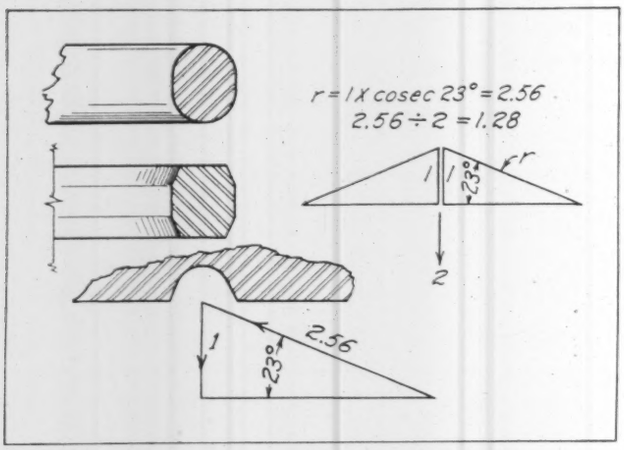

Fig. 1—Force diagram, standard ring type joint

pressure autoclave. Thereupon he computed the bolt load due to internal pressure. After some thought, he added certain large amounts to this load in order to prevent any slipping away of pressure, made provision for extremely heavy flanges studded with alloy steel bolts, checked his computations, and sketched away contentedly.

Manifestly, the implications of this story are without justification, unless supplemented by the presentation of factual data relating to improved designs and construction. Accordingly I shall describe a number of types which indicate that the usual bolted joint construction is being largely modified or superseded by more reliable, more easily maintained, and more economical forms.

Two forms of joints are in common use: The boiler manhole type and the ring type.

Manhole Type Joints

The manhole, or "internal cover," type is applicable to and superior for many kinds of apparatus; for example, certain autoclaves and also feedwater heaters, especially when high temperature, high pressure services are in question. Maintaining large bolted joints is a serious problem, as the rapid introduction of heat causes differential expansion between flanges and bolting. On the other hand if the manhole type is used, increasing internal pressure automatically increases the gasket pressure and thus the internal pressure assists in sealing; so that all temperature swings will fail to impair the inherent reliability of the joint. Nevertheless, in certain instances, manhole and handhole plates, fitted with solid metal gaskets, have failed in service due to faulty design. Bolts for such covers have two functions: (1) they must be adequate to "bed" or cold flow the gasket sufficiently to assure a tight joint for the initial hydrostatic test; (2) they serve as fastenings in assembly. Bolting which will set a woven asbestos or a steel-asbestos gasket, such as commonly used for low or medium pressure boilers, will make but slight impression upon the solid metal gaskets which are frequently specified for high pressure boilers (say 900 psi or higher), particularly when gaskets are wide.

Designers should not overlook this useful type of closure; and they should provide bolting, yoke or "crab claw," and gasket which are correct according to engineering principles and not selected in a haphazard manner from stock or a catalog.

Ring Type Joints

Ring type joints probably originated on the Rand, where they were used on rising mains for water supply. They are now a standby in oil fields as well as in petroleum refining; moreover, their use is expanding steadily both in high temperature, high pressure steam plants and in various industries.

The usual oval and octagonal cross-section rings are now a standardized product, as witness the API Standards on Ring Joints for Steel Flanges and Flange Unions No. 5-G-3-1937 (which include 150, 300, 400, 600, 900, 1500 and 2500 lb flanges) and the American Standard for Steel Pipe Flanges and Flanged Fittings, B16e-1939. A 4000 lb standard is now proposed[1] in which flange dimensions for 4000 lb, 900 F flanges and flanged fittings are "based on the ring joint, and this type of joint is primarily recommended for the 4000 lb standard."

A comparison of the tables of Service Pressure Ratings for Carbon Steel Flanges and Flanged Fittings to confine water, steam or oil, as given in ASA B16e-1939, shows that within the range of service temperatures below that for the basic or "primary service pressure rating," higher pressures are allowed for ring joint facings than for standard facings. Furthermore, the tables of Service Pressure Ratings for Carbon-Molybdenum and Equivalent Alloy Steels at Temperatures of 1000 F and Below show "primary service pressure ratings" set at 900 F for standard facings, but at 950 F for ring joint facings. At a service temperature of 100 F the ratings for ring joint facings are 20 per cent higher than for standard facings; they are also higher, in proper proportion, throughout the temperature range between 100 F and 1000 F.

[1]Standardization of Flanged Steel Products, Including Proposed 4000-Lb Standard, by A. M. Houser, *Valve World*, May-June, 1939.

The usual bolted joint construction is being largely modified or superseded by more reliable, more easily maintained, and more economical forms. The author describes several of these current types of gaskets and closures

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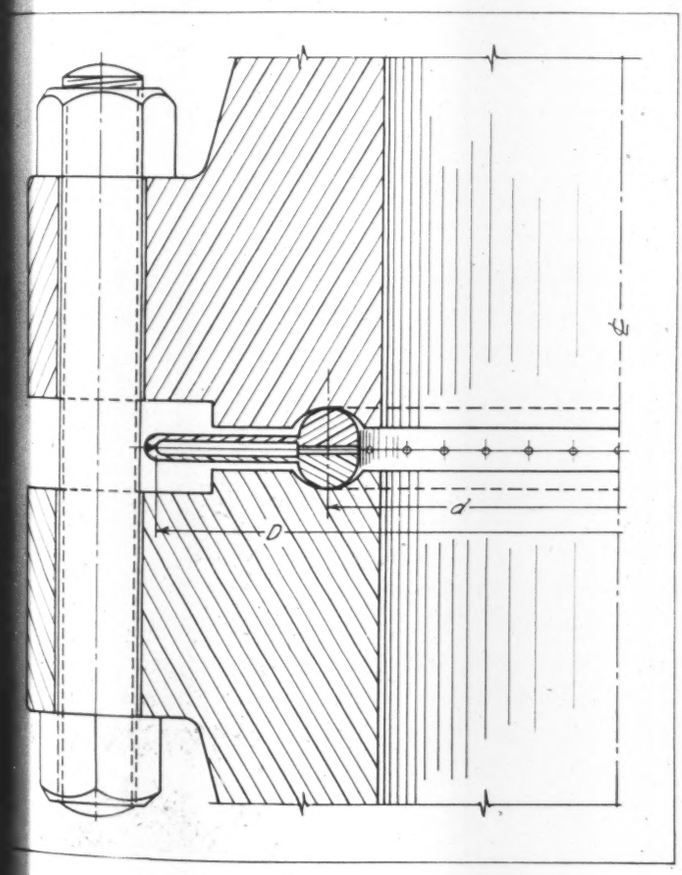
Two methods (smooth forging and machining) have been used to manufacture rings. They have been smooth forged out of various kinds of iron without any machine finish whatever. Sometimes after injury in service they have been reconditioned by cold working, i. e., squeezing between dies in a press or by "striking up" between drop hammer dies.

Machined rings are made of various grades of steel and such rings *possess a particular advantage over nearly all other gaskets.* The advantage is that the finely finished and accurate surfaces of both ring and groove provide the perfectly matching surfaces which will insure a tight joint with minimum gasket pressure. Fig. 1, drawn to scale, shows that the pressure on one face of the ring[2] is equal to about 1¼ times the flange pressure. Tests[3] have proved that for any given working pressure the ring joint requires less bolt load, and consequently imposes lower flange stresses than those required for similar sizes of joints with flat gaskets. Under heavy internal fluid pressures the ring tends to expand radially, and thus to a certain extent it has self-sealing properties.

[2]Gaskets for Pressure Vessels and Heat Exchangers, by C. O. Sandstrom, *Chemical and Metallurgical Engineering*, March, 1934, p. 134.
[3]The Ring Joint—Its Relative Merit and Application, by E. C. Petrie, Heating, Piping and Air Conditioning, April, 1937, pp. 213-220.

Fig. 2—Bellows gasket



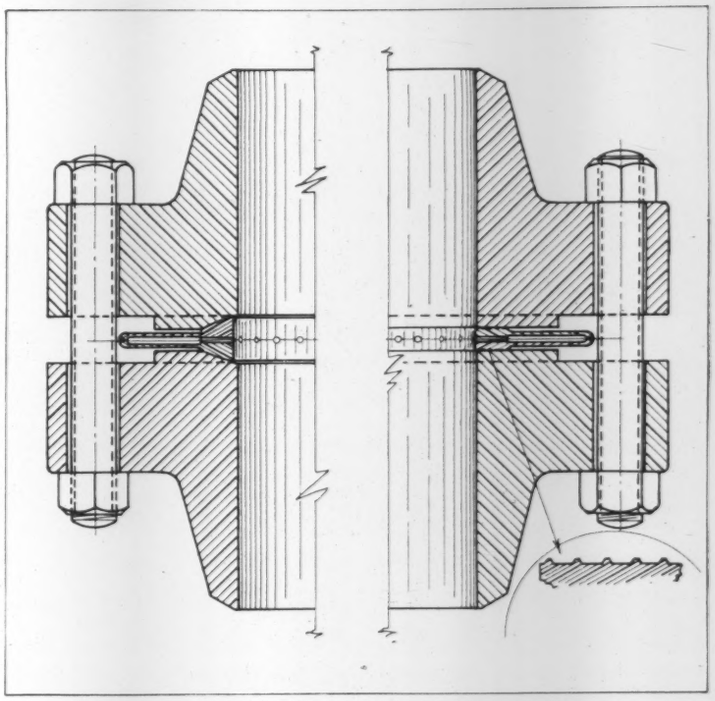


Fig. 3 (left)—Bellows gasket with lens ring . . . Fig. 4 (right)—Bellows gasket with solid profile ring

Refinery specifications frequently call for calorized rings, i. e., rings with an iron-aluminum surface alloy about 0.01 in. thick, resistant to oxidation and sulphur corrosion, and sufficiently ductile, due to the diffusion of the aluminum alloy into the base steel, to be safe against injury to the protective surface when the ring is clamped between flanges. In some of the more recent specifications, however, 18-8 stainless steel has superseded the calorized rings.

There does not appear to be any reason for a limit to the size of ring joints. I have seen rings with a cross-section 2 in. high and 1¼ in. wide, and know of rings 100 in. in diameter.

The Bellows Type Joint

In Europe a type of gasket new to us and known as the "bellows type" has been developed to overcome joint troubles whether due to differential expansion caused by rapid swings of temperature or to creep stresses in flanges and bolts. It is said that this type was originated by the German dye industry and has also been used in German chemical plants. It was illustrated in the First Report of the Pipe Flanges Research Committee,[4] which states that the ring should be so designed that a contact pressure at least equal to and dependent solely on the internal pressure, should be the optimum sought, with the result that "it would not be necessary to store reserve elastic strain in the bolts, therefore the initial stress in the bolts could be very considerably reduced." It was further stated: "There was nothing novel in this form of joint ring, it was simply another application of an old idea . . ."

Figs. 2, 3 and 4 illustrate, respectively, ring, lens, and

[*Concluded on page* 506]

[4]Published by the Institution of Mechanical Engineers, Storey's Gate, St. Jame's Park, London, S. W. 1, February, 1936.

The Heating Practices of Consulting Engineers

By Samuel R. Lewis*

THE practice of engineering, in its final analysis, is seeking the truth. The practice of consulting engineering in this office may be resolved into the procedure of determining the facts concerning a project and then in making the prescription fit the facts.

The minimum equipment required for the practice of engineering consists of a knowledge of methods and materials with their ranges of usefulness and their limitations, an appreciation of economics, and an application of judgment—sometimes referred to as "horse sense." These same general requirements may be specified as essentials to the conduct of any business, but in the practice of engineering they have a particular importance. The independent engineer who adapts the project and the instruments of service each to the other performs a useful service, the benefits of which accrue both to the owner and to the manufacturer.

Recently we were asked by the authorities in West Virginia to advise whether unit ventilating machines or central fan heating systems should be used for some 15 different new school buildings to be built throughout a county. It appears that in West Virginia (which always has been unconventionally independent anyway) they have eliminated the separate school board of horse and buggy and mud road days for every little school house.

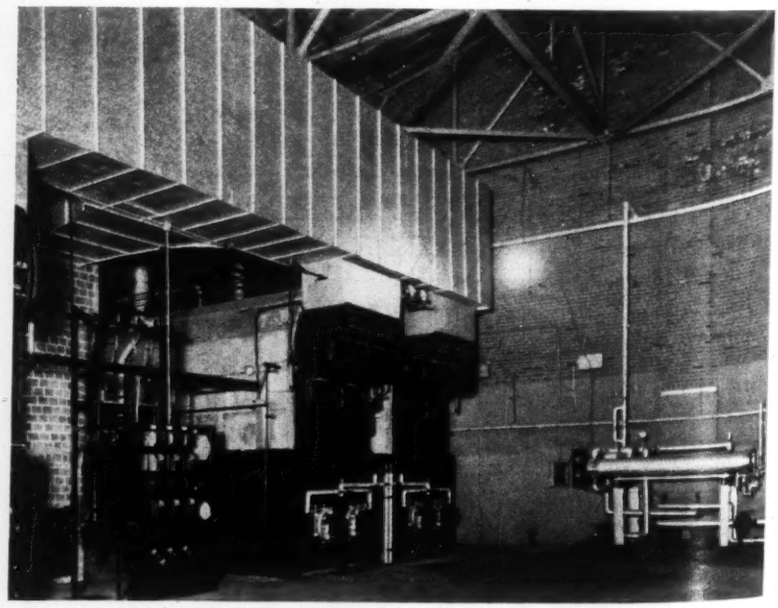
Our advice was that no single answer could possibly be given, but that the type of equipment for each separate school must be decided on its merits. The type of fuel available, the sort of janitor or fireman service, the possibility of future additions, the height above the ground of the first story, and several other items, all aid in arriving at a decision as to which type of plant would be best for each building.

One large state university has forced hot water heating from its central plant, the circulated hot water acting as a condensing medium for the exhaust steam from the turbines which drive the electric generators. Therefore, we designed the heating plants for the new buildings to employ hot water. Another large state university distributes exhaust steam to the various buildings, with electric vacuum pumps in each building and with condensate return pipes thus under pressure. Therefore, we must use steam in these buildings. In another case, the only available water for steam making purposes is objectionable because of extreme hardness. Since there was no particular need for steam we used forced hot water, with special baffling arrangements within the very large oil burning boilers to insure mechanically induced circulation of the water over the heat absorbing surfaces. The results in efficiency were surprisingly good, involving a saving in fuel in a very large, completely ventilated school of well over 16 per cent as compared with a duplicate steam heated school in the same city during the same season. We are observing with great interest the comparative life of boiler tubes, return pipes and the like.

In another case of a complete air conditioning system, the copper finned convectors first touched by the entering air necessarily must have a very high heat transmitting efficiency in summer, when the well water used is only a few degrees colder than the air. These convectors would freeze too easily and quickly in winter, since the waterways are very small. This situation was met more or less satisfactorily by circulating water in winter carrying a percentage of ethylene glycol, and by storing this liquor in barrels during the summer while the well water used for cooling was wasted.

A similar plant, designed more recently, was arranged for reversing the water circulation by manipulation of valves near the pump, so that in winter the warmest water enters the side of the convectors which is touched by the coolest entering air and so that in summer the coolest water enters the leeward side of the convectors.

Occasionally we meet unusual conditions, as in buildings below flood water level on the Ohio River. This situation has been met successfully by placing the oil or gas burning boilers and heating mains in the top stories or attics of buildings, with easily removal unit heaters for the lower stories. Forced circulation hot water lends itself admirably to this sort of service. Unit heaters, using either steam or hot water or direct fired



The oil burning boilers at a manufacturing plant in La Crosse, Wis.

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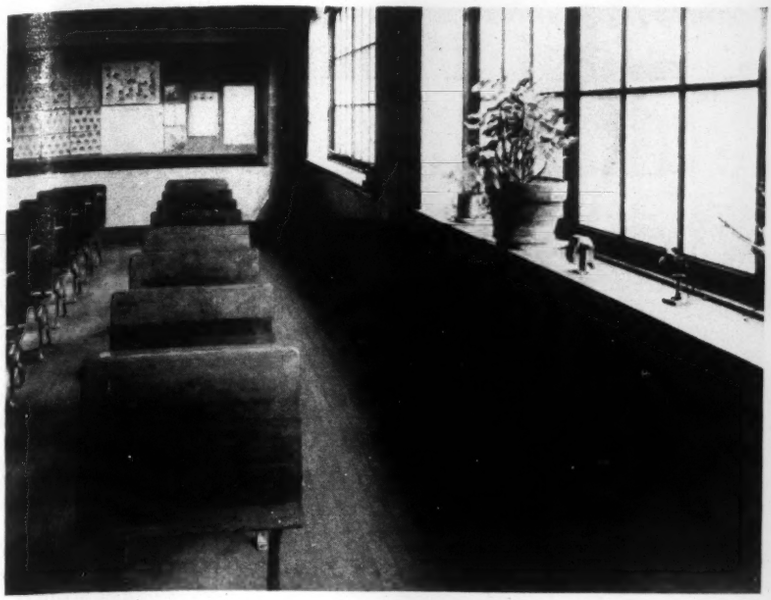


Warmed air in introduced to this classroom through slots along the floor under the windows. This scheme is typical of some 50 rooms in the Toledo, Ohio, schools and a museum

gas, practically have superseded pipes or radiators for heating conventional industrial plants.

Our practice in approaching the heating or cooling of any building is to bring to bear a considerable pressure on the owner to make the structure easy and economical to heat or to cool. Sometimes this leads to interesting results, though they may be embarrassing. For instance, for a large elevator manufacturing plant, formerly equipped with coal burning unit heaters scattered around between the machines, we estimated that a forced hot water unit heating system with boiler house and chimney would cost $40,000. If the roof and certain walls were to be insulated, the heating plant would cost $8,000 less and the insulation would cost $7,000. The annual fuel cost, with the building uninsulated, would be $4,000, dropping to $2,700 with insulation. Thus the insulation would be paid for in a few years by the fuel saved.

We found that natural gas was available. With gas burning unit heaters no boiler house or chimney are needed, the piping is much smaller, and no pipe insulation is required. The gas burning plant therefore was installed, at an investment price of $10,000 with complete automatic control, to maintain 50 F. We estimated that the annual cost of gas, at 50 cents per thousand cubic feet, would be around $6,000—justifiable on account of the accompanying reduction in investment for heating and ventilation from about $40,000 to $17,000 ($10,000 for heating and $7,000 for insulating the building).

It is proposed to publish, from time to time, brief discussions of the heating practices of representative consulting engineers. In its final analysis, the practice of engineering is seeking the truth and the practice of consulting engineering may be resolved into the procedure of determining the facts concerning a project and then making the prescription fit the facts. . . . It is the aim of these informal discussions to show how such prescriptions have been written

I quote from a letter from the owner, dated May 2, 1939: "Our meter readings for the entire heating season up to April 21, 1939, at the price of 50 cents per thousand for gas, give a fuel cost of $2,066. For the last six weeks the weather has been cooler than normal and I am inclined to think that year in and year out our fuel cost will run from $2,000 to $2,500.

"It is far ahead of anything that we anticipated in cleanliness, and the monetary value of the reduction in noise due to the ceiling insulation and the absence of dust and dirt is impossible to estimate."

Evidently the practice in this office of estimating the efficiency of gas unit heaters must be revised, or we under-estimated the care which would be exercised by the owner, or we did not sufficiently evaluate the benefit of the insulation. We were embarrassed by having the annual fuel cost of $2,500 against our estimate of $6,000, but we are glad to bear red faces when our clients' faces shine.

Gasification of Coal

High pressures may someday, through research, be used on a large scale by the manufactured gas and coal industries of the country, to their mutual advantage. Such is the prediction made in a technical report of Bituminous Coal Research, Inc., on Possibilities of Research in the Gasification of Coal, by C. A. Barnes, fuel engineer at Battelle Memorial Institute. According to the report, statistics show that, despite the widespread distribution of natural gas in recent years, the present trend is toward an increased demand for manufactured gas.

Dr. Barnes states that improved gasification processes, possibly employing high pressures to obtain a gas of high calorific value without the use of oil for enrichment, are of equal interest to the gas utility and the coal producer, and research on the subject merits their mutual support. That the prediction made in the report is no "pipe dream" is evidenced by recent reports from England which state that ". . . results obtained indicate the possibility of gasifying at these high pressures a large proportion of what has been regarded as the fixed carbon of coal in the form of a gas rich in methane and of high calorific value."

Promote Use of Coke

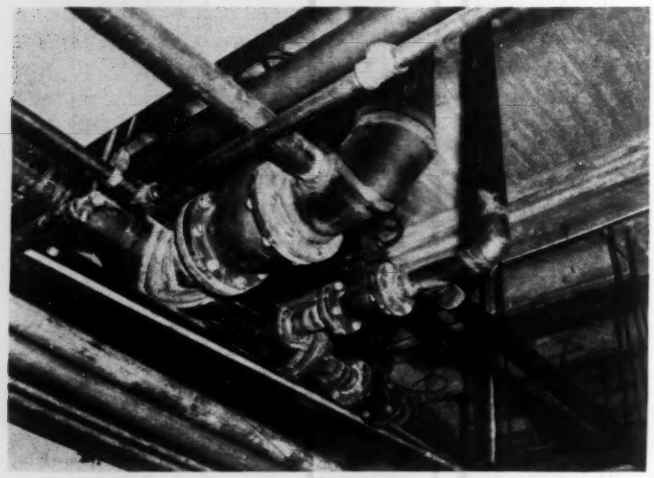
From the standpoint of gross consumption of coal, processes by which gas is made by carbonization of coal are more favorable to the coal industry than processes of complete gasification, but the production of coal gas has often been limited by the inability to dispose of the coke. For this reason, the report says, the coal industry should coöperate with the manufactured gas industry to promote the use of coke.

Welding Assures Tight Piping System for Moody Bible Institute's Building

By J. A. Hudson*

OXY-ACETYLENE welding assures permanent tightness of the 1985 ft of piping for the vacuum type steam heating system in the new 12 story administration building of the Moody Bible Institute, Chicago.

Steam for the system is supplied as exhaust from 14 in. exhaust headers in the engine room. There the steam is originally produced at 200 lb per sq in. and used for generating electricity. The exhaust steam is piped through a tunnel to the basement of the administration building one-half block away. Here an 8 in. line rises to the 8th floor ceiling from which point the steam is distributed throughout the building by means of a combination down- and upfeed vacuum system.



Detail of distribution system in the 8th floor. Fifteen field welds are included in this section of piping

Fabricating Fittings

Since the piping was of a nature which readily lent itself to complete assembly and installation on the job, its layout was designed on that basis. Most of the special fittings and bends were designed with mitered joints and the templets necessary for this work were all carefully laid out and prepared before work was started.

Although machine beveled piping was used throughout the system, all fittings were fabricated and beveled by means of the oxy-acetylene cutting blowpipe. The parts for mitered turns and bends were cut from standard pipe. Once the chalk mark was made on the pipe, with the proper templet as a guide, it was a simple matter to shape the part with the cutting blowpipe. In each case, a true mitered cut was produced by holding the blowpipe flame directly in the plane of cut. After beveling with the blowpipe, the parts were welded together so that the entire turn or bend was completed before being incorporated in a unit section.

Unit sections of a size suitable for convenient handling were assembled by welding close by the points of installation. These sections were each raised into position and tie-in welds were made. This procedure speeded erection and permitted the amount of scaffolding required to be reduced to a minimum. In fact, many of the tie-in welds were made from a ladder.

Pipe anchors and supports, as needed, were also fabricated with the oxy-acetylene blowpipe on the job. Two of these supports can be seen close to the left-hand border of the accompanying illustration. With a wide variety of designs from which to choose, it was possible to make each individual anchor and support as strong and rigid as its location demanded, using a minimum of metal and a minimum of time. Many odd lengths of scrap pipe were used for this purpose.

All of the 443 welds required in installing this heating system were made on various diameters of pipe, as follows:

Pipe Diam., In.	Pipe Lengths, Ft	No. of Welds
14	20	7
10	210	23
8	140	27
6	220	28
5	190	26
4	140	54
3½	75	42
3	250	84
2½	465	62
2	275	91

Because of corrosive conditions, wrought iron was used for the hot and cold water lines, cold water storage tank, and surge tank.

Thielbar and Fugard were the architects, and the contractors were Thos. J. Douglass & Co.

Copper Mine Extends Air Conditioning

Extension to the 4000 ft level of the air conditioning system which made the Magma Copper mine, Superior, Ariz., the first mechanically air conditioned mine in the United States, has been announced. The previous air conditioning extended to the 3600 ft level.

Centrifugal type refrigerating machines will be used for the cooling. Water will be taken from condensers now on the job and sprayed through a tunnel. Foul air from the mine then will be taken from an upcast shaft and blown through this fine spray to cool the water. This air cooled condenser water is returned to the collecting tank and used for condensing in the two centrifugal refrigerating machines installed on the 3600 ft level.

*Supervisor of Process Service, The Linde Air Products Co.

Properties of Mixtures of Air and Saturated Water Vapor for Barometric Pressures from 22 In. to 32 In. of Mercury

By William Goodman*

Barometric Pressure 23 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)		Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu		Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s'		h_v	h	t	w	w	v_D		s'		h_v	h
1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
−40	0.0001514	1.060	13.76	0.02434	0.2401	0.1606	0.1579	−9.442	40	0.006778	47.45	16.39	1.089	0.2431	7.192	7.306	16.91
−39	.0001603	1.122	13.79	.02577	.2401	.1700	.1672	−9.193	41	.007048	49.34	16.42	1.133	.2432	7.478	7.610	17.45
−38	.0001698	1.189	13.83	.02730	.2401	.1802	.1773	−8.943	42	.007330	51.31	16.45	1.178	.2433	7.777	7.918	18.00
−37	.0001798	1.258	13.86	.02890	.2401	.1907	.1878	−8.692	43	.007619	53.34	16.48	1.225	.2434	8.084	8.233	18.55
−36	.0001903	1.332	13.89	.03059	.2401	.2019	.1988	−8.441	44	.007920	55.44	16.52	1.273	.2436	8.403	8.561	19.12
−35	0.0002012	1.408	13.93	0.03285	0.2401	0.2135	0.2103	−8.190	45	0.008236	57.65	16.55	1.323	0.2437	8.738	8.907	19.71
−34	.0002129	1.491	13.96	.03423	.2401	.2259	.2227	−7.937	46	.008558	59.91	16.58	1.375	.2439	9.080	9.259	20.30
−33	.0002251	1.576	13.99	.03619	.2401	.2388	.2355	−7.685	47	.008891	62.24	16.61	1.429	.2440	9.434	9.624	20.90
−32	.0002381	1.666	14.02	.03827	.2401	.2526	.2492	−7.431	48	.009239	64.67	16.65	1.485	.2442	9.802	10.00	21.52
−31	.0002514	1.760	14.06	.04042	.2401	.2667	.2632	−7.177	49	.009596	67.18	16.68	1.542	.2443	10.18	10.40	22.16
−30	0.0002658	1.861	14.09	0.04273	0.2401	0.2820	0.2784	−6.922	50	0.009970	69.79	16.71	1.602	0.2445	10.58	10.81	22.81
−29	.0002807	1.965	14.12	.04513	.2401	.2978	.2942	−6.666	51	.01036	72.50	16.75	1.664	.2447	10.99	11.23	23.47
−28	.0002962	2.073	14.16	.04761	.2401	.3142	.3105	−6.410	52	.01075	75.24	16.78	1.728	.2448	11.40	11.66	24.14
−27	.0003128	2.190	14.19	.05029	.2401	.3319	.3281	−6.152	53	.01117	78.16	16.81	1.794	.2450	11.85	12.11	24.83
−26	.0003299	2.310	14.22	.05304	.2402	.3501	.3462	−5.894	54	.01159	81.10	16.84	1.861	.2452	12.29	12.57	25.53
−25	0.0003480	2.436	14.25	0.05595	0.2402	0.3692	0.3653	−5.635	55	0.01202	84.17	16.88	1.932	0.2454	12.76	13.06	26.26
−24	.0003671	2.570	14.29	.05902	.2402	.3895	.3855	−5.375	56	.01248	87.36	16.91	2.004	.2456	13.24	13.56	27.00
−23	.0003872	2.711	14.32	.06225	.2402	.4108	.4068	−5.113	57	.01295	90.62	16.94	2.080	.2458	13.74	14.07	27.75
−22	.0004080	2.856	14.35	.06559	.2402	.4329	.4289	−4.851	58	.01344	94.05	16.98	2.158	.2461	14.26	14.61	28.53
−21	.0004299	3.009	14.39	.06911	.2402	.4561	.4521	−4.588	59	.01393	97.52	17.01	2.238	.2463	14.78	15.15	29.31
−20	0.0004530	3.171	14.42	0.07283	0.2402	0.4806	0.4765	−4.324	60	0.01445	101.1	17.04	2.321	0.2465	15.33	15.72	30.12
−19	.0004772	3.341	14.45	.07672	.2402	.5063	.5023	−4.058	61	.01498	104.9	17.07	2.407	.2467	15.90	16.31	30.95
−18	.0005023	3.516	14.48	.08075	.2402	.5329	.5288	−3.791	62	.01554	108.8	17.11	2.496	.2470	16.49	16.93	31.81
−17	.0005286	3.700	14.52	.08499	.2402	.5609	.5569	−3.523	63	.01612	112.8	17.14	2.588	.2473	17.10	17.56	32.68
−16	.0005566	3.896	14.55	.08949	.2403	.5906	.5866	−3.253	64	.01670	116.9	17.17	2.683	.2475	17.72	18.20	33.56
−15	0.0005853	4.097	14.58	0.09410	0.2403	0.6210	0.6170	−2.983	65	0.01732	121.2	17.21	2.780	0.2478	18.37	18.88	34.48
−14	.0006157	4.310	14.61	.09898	.2403	.6532	.6493	−2.711	66	.01794	125.6	17.24	2.881	.2481	19.03	19.56	35.40
−13	.0006476	4.533	14.65	.1041	.2403	.6871	.6834	−2.437	67	.01860	130.2	17.27	2.987	.2484	19.73	20.29	36.37
−12	.0006807	4.765	14.68	.1094	.2403	.7223	.7186	−2.161	68	.01927	134.9	17.30	3.094	.2487	20.44	21.03	37.35
−11	.0007150	5.005	14.71	.1149	.2403	.7586	.7551	−1.885	69	.01996	139.7	17.34	3.205	.2490	21.18	21.79	38.35
−10	0.0007520	5.264	14.75	0.1209	0.2403	0.7978	0.7944	−1.606	70	0.02068	144.8	17.37	3.321	0.2493	21.94	22.59	39.39
−9	.0007895	5.526	14.78	.1269	.2404	.8376	.8345	−1.326	71	.02141	149.9	17.40	3.439	.2496	22.72	23.40	40.44
−8	.0008298	5.808	14.81	.1334	.2404	.8804	.8774	−1.043	72	.02218	155.3	17.43	3.562	.2500	23.54	24.25	41.53
−7	.0008705	6.094	14.84	.1400	.2404	.9236	.9209	−0.7591	73	.02297	160.8	17.47	3.689	.2503	24.38	25.12	42.64
−6	.0009137	6.396	14.88	.1469	.2404	.9694	.9670	−0.4730	74	.02379	166.5	17.50	3.820	.2507	25.24	26.03	43.79
−5	0.0009595	6.717	14.91	0.1543	0.2404	1.018	1.016	−0.1841	75	0.02463	172.4	17.53	3.954	0.2511	26.13	26.95	44.95
−4	.001007	7.046	14.94	.1618	.2405	1.068	1.066	+0.1061	76	.02549	178.5	17.57	4.093	.2515	27.05	27.91	46.15
−3	.001056	7.390	14.98	.1697	.2405	1.120	1.119	0.3987	77	.02641	184.8	17.60	4.239	.2519	28.02	28.92	47.40
−2	.001108	7.753	15.01	.1781	.2405	1.175	1.174	0.6942	78	.02733	191.3	17.63	4.388	.2523	28.99	29.95	48.67
−1	.001162	8.132	15.04	.1868	.2405	1.233	1.232	0.9921	79	.02828	198.0	17.66	4.541	.2527	30.01	31.00	49.96
0	0.001217	8.519	15.07	0.1957	0.2406	1.291	1.291	1.291	80	0.02926	204.8	17.70	4.698	0.2532	31.05	32.09	51.29
1	.001275	8.925	15.11	.2050	.2406	1.353	1.353	1.593	81	.03029	212.0	17.73	4.863	.2536	32.14	33.23	52.67
2	.001336	9.351	15.14	.2148	.2406	1.417	1.419	1.899	82	.03134	219.4	17.76	5.030	.2541	33.25	34.39	54.07
3	.001400	9.797	15.17	.2250	.2406	1.485	1.487	2.207	83	.03243	227.0	17.80	5.205	.2546	34.41	35.60	55.52
4	.001466	10.27	15.20	.2358	.2407	1.556	1.559	2.519	84	.03353	234.7	17.83	5.383	.2551	35.58	36.83	56.99
5	0.001535	10.75	15.24	0.2468	0.2407	1.629	1.632	2.832	85	0.03469	242.8	17.86	5.568	0.2556	36.80	38.12	58.52
6	.001606	11.25	15.27	.2583	.2407	1.704	1.709	3.149	86	.03589	251.2	17.89	5.760	.2562	38.07	39.45	60.09
7	.001681	11.77	15.30	.2703	.2408	1.784	1.789	3.469	87	.03712	259.8	17.93	5.956	.2567	39.38	40.82	61.70
8	.001759	12.31	15.34	.2828	.2408	1.867	1.873	3.793	88	.03838	268.6	17.96	6.159	.2573	40.72	42.22	63.34
9	.001841	12.89	15.37	.2960	.2408	1.953	1.961	4.121	89	.03970	277.9	17.99	6.371	.2579	42.12	43.69	65.05
10	0.001925	13.47	15.40	0.3095	0.2409	2.042	2.051	4.451	90	0.04105	287.4	18.02	6.587	0.2585	43.55	45.19	66.79
11	.002015	14.10	15.43	.3239	.2409	2.137	2.148	4.788	91	.04245	297.2	18.06	6.812	.2591	45.04	46.76	68.60
12	.002106	14.74	15.47	.3386	.2410	2.235	2.246	5.126	92	.04388	307.1	18.09	7.041	.2597	46.55	48.34	70.42
13	.002199	15.39	15.50	.3536	.2410	2.333	2.346	5.466	93	.04537	317.6	18.12	7.280	.2604	48.14	50.01	72.33
14	.002301	16.11	15.53	.3699	.2410	2.441	2.456	5.816	94	.04691	328.4	18.16	7.526	.2611	49.77	51.72	74.28
15	0.002404	16.83	15.57	0.3865	0.2411	2.551	2.567	6.167	95	0.04849	339.4	18.19	7.779	0.2618	51.44	53.49	76.29
16	.002510	17.57	15.60	.4036	.2411	2.664	2.682	6.522	96	.05011	350.8	18.22	8.040	.2626	53.17	55.30	78.34
17	.002621	18.35	15.63	.4214	.2412	2.781	2.802	6.882	97	.05178	362.5	18.25	8.308	.2633	54.94	57.16	80.44
18	.002737	19.16	15.66	.4401	.2412	2.904	2.927	7.247	98	.05354	374.8	18.29	8.589	.2641	56.81	59.13	82.65
19	.002859	20.01	15.70	.4596	.2413	3.033	3.058	7.618	99	.05532	387.2	18.32	8.874	.2649	58.69	61.11	84.87
20	0.002983	20.88	15.73	0.4795	0.2413	3.164	3.191	7.991	100	0.05718	400.2	18.35	9.172	0.2657	60.66	63.19	87.19
21	.003114	21.80	15.76	.5007	.2414	3.304	3.334	8.374	101	.05909	413.6	18.39	9.479	.2666	62.69	65.33	89.57
22	.003248	22.74	15.79	.5222	.2415	3.446	3.479	8.759	102	.06107	427.5	18.42	9.794	.2675	64.80	67.55	92.03
23	.003389	23.72	15.83	.5449	.2415	3.596	3.631	9.151	103	.06311	441.8	18.45	10.12	.2684	66.96	69.83	94.55
24	.003535	24.75	15.86	.5683	.2416	3.751	3.789	9.549	104	.06522	456.5	18.48	10.46	.2694	69.19	72.19	97.15
25	0.003684	25.79	15.89	0.5923	0.2417	3.909	3.951	9.951	105	0.06737	471.6	18.52	10.80	0.2703	71.48	74.60	99.80
26	.003844	26.91	15.93	.6180	.2417	4.078	4.123	10.36	106	.06961	487.3	18.55	11.16	.2713	73.86	77.12	102.56
27	.004005	28.04	15.96	.6440	.2418	4.250	4.299	10.78	107	.07195	503.6	18.58	11.54	.2724	76.33	79.73	105.41
28	.004174	29.22	15.99	.6710	.2419	4.429	4.481	11.20	108	.07433	520.3	18.62	11.92	.2735	78.86	82.40	108.32
29	.004350	30.45	16.02	.6993	.2420	4.615	4.672	11.63	109	.07680	537.6	18.65	12.31	.2746	81.48	85.17	111.33
30	0.004532	31.73	16.06	0.7287	0.2420	4.809	4.870	12.07	110	0.07935	555.5	18.68	12.72	0.2757	84.20	88.04	114.44
31	.004723	33.06	16.09	.7594	.2421	5.012	5.078	12.52	111	.08196	573.7	18.71	13.14	.2769	86.96	90.97	117.61
32	.004915	34.41	16.12	.7901	.2422	5.215	5.288	12.9	112	.08469	592.8	18.75	13.58	.2781	89.86	94.03	120.91
33	.005123	35.86	16.16	.8233	.2423	5.435	5.513	13.43	113	.08751	612.6	18.78	14.03	.2794	92.85	97.20	124.32
34	.005335	37.34	16.19	.8575	.2424	5.660	5.744	13.90	114	.09041	632.9	18.81	14.49	.2807	95.93	100.46	127.82
35	0.005553	38.87	16.22	0.8926	0.2425	5.891	5.981	14.38	115	0.09340	653.8	18.84	14.97	0.2820	99.10	103.83	131.43
36	.005782	40.47	16.25	0.9294	.2426	6.135	6.231	14.87	116	.09653	675.7	18.88	15.47	.2834	102.42	107.34	135.18
37	.006019	42.13	16.29	0.9671	.2427	6.386	6.488	15.37	117	.09974	698.2	18.91	15.98	.2849	105.82	110.95	139.03
38	.006263	43.84	16.32	1.007	.2428	6.645	6.754	15.87	118	.1030	721.3	18.94	16.51	.2864	109.33	114.66	142.98
39	.006516	45.61	16.35	1.047	.2429	6.913	7.030	16.39	119	.1064	745.0	18.98	17.06	.2879	112.92	118.48	147.04

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(The tabulation for 22 in. barometric pressure was published in the July issue.)

Properties of Mixtures of Air and Saturated Water Vapor—(Continued)
Barometric Pressure 23 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)		Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu		Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s'		h_v	h	t	w	w	v_D		s'		h_v	h
1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
120	0.1100	770.0	19.01	17.63	0.2895	116.71	122.51	151.31	160	0.4529	3170.5	20.32	72.30	0.4438	480.56	511.90	550.30
121	.1137	795.6	19.04	18.21	.2912	120.59	126.63	155.67	161	.4721	3304.5	20.35	75.34	.4524	500.87	533.72	572.36
122	.1174	821.9	19.07	18.81	.2928	124.57	130.85	160.13	162	.4925	3447.2	20.39	78.59	.4616	522.49	556.96	595.54
123	.1214	849.5	19.11	19.44	.2946	128.75	135.29	164.81	163	.5139	3597.1	20.42	81.99	.4712	545.22	581.44	620.56
124	.1254	877.8	19.14	20.09	.2964	133.05	139.86	169.62	164	.5367	3757.1	20.45	85.64	.4815	569.47	607.47	646.83
125	0.1296	907.2	19.17	20.76	0.2983	137.51	144.61	174.61	165	0.5609	3926.2	20.48	89.48	0.4924	595.09	635.03	674.63
126	.1340	937.7	19.21	21.45	.3003	142.12	149.52	179.76	166	.5868	4107.6	20.52	93.60	.5041	622.59	664.61	704.45
127	.1385	969.2	19.24	22.17	.3023	146.91	154.60	185.08	167	.6142	4299.1	20.55	97.95	.5164	651.62	695.84	735.92
128	.1431	1002.0	19.27	22.92	.3044	151.87	159.89	190.61	168	.6436	4505.5	20.58	102.6	.5296	682.90	729.50	769.82
129	.1480	1035.8	19.30	23.70	.3066	157.00	165.34	196.30	169	.6748	4723.8	20.62	107.6	.5437	715.99	765.12	805.68
130	0.1531	1071.6	19.34	24.50	0.3089	162.42	171.13	202.33	170	0.7085	4959.6	20.65	113.0	0.5588	751.73	803.59	844.39
131	.1583	1107.8	19.37	25.33	.3112	167.91	176.98	208.42	171	.7444	5210.9	20.68	118.7	.5750	789.83	844.62	885.66
132	.1635	1144.6	19.40	26.18	.3136	173.50	182.93	214.61	172	.7834	5483.6	20.71	124.9	.5925	831.16	889.13	930.41
133	.1692	1184.1	19.43	27.08	.3161	179.47	189.30	221.22	173	.8253	5776.9	20.75	131.5	.6114	875.61	937.01	978.53
134	.1750	1224.9	19.47	28.00	.3187	185.65	195.89	228.05	174	.8708	6095.3	20.78	138.7	.6318	923.88	989.01	1030.8
135	0.1810	1266.8	19.50	28.97	0.3214	192.01	202.67	235.07	175	0.9199	6439.2	20.81	146.5	0.6540	976.00	1045.2	1087.2
136	.1872	1310.3	19.53	29.96	.3242	198.61	209.71	242.35	176	0.9736	6815.5	20.84	155.1	.6781	1033.0	1106.6	1148.9
137	.1937	1355.9	19.57	30.99	.3272	205.52	217.10	249.98	177	1.032	7185.4	20.88	164.4	.7045	1095.2	1173.6	1216.1
138	.2006	1403.9	19.60	32.08	.3303	212.78	224.86	257.98	178	1.097	7675.5	20.91	174.6	.7334	1163.4	1247.2	1289.9
139	.2075	1452.6	19.63	33.19	.3334	220.17	232.74	266.10	179	1.167	8171.8	20.94	185.9	.7653	1238.6	1328.2	1371.1
140	0.2148	1503.5	19.66	34.35	0.3367	227.88	240.98	274.58	180	1.246	8,721.3	20.98	198.4	0.8007	1321.9	1418.0	1461.2
141	.2223	1556.0	19.70	35.56	.3400	235.85	249.50	283.34	181	1.333	9,331.7	21.01	212.2	.8399	1414.4	1517.7	1561.2
142	.2303	1612.3	19.73	36.83	.3437	244.38	258.62	292.70	182	1.431	10,016	21.04	227.7	.8839	1518.2	1629.7	1673.5
143	.2384	1668.9	19.76	38.14	.3473	252.96	267.79	302.11	183	1.541	10,786	21.07	245.2	.9334	1634.8	1755.4	1799.4
144	.2470	1728.8	19.80	39.50	.3511	262.04	277.48	312.04	184	1.666	11,661	21.11	265.0	.9897	1767.5	1898.6	1943.5
145	0.2559	1791.3	19.83	40.93	0.3552	271.51	287.66	322.46	185	1.809	12,662	21.14	287.7	1.054	1919.2	2062.3	2106.7
146	.2653	1857.3	19.86	42.42	.3594	281.52	298.36	333.40	186	1.974	13,816	21.17	313.9	1.128	2094.1	2251.0	2295.6
147	.2750	1925.3	19.89	43.96	.3638	291.82	309.39	344.67	187	2.166	15,163	21.21	344.5	1.215	2298.2	2471.3	2516.2
148	.2852	1996.5	19.93	45.58	.3684	302.62	320.96	356.48	188	2.394	16,755	21.24	380.6	1.317	2539.5	2731.7	2776.8
149	.2958	2070.5	19.96	47.27	.3731	313.83	332.97	368.73	189	2.667	18,672	21.27	424.0	1.440	2830.1	3045.4	3090.7
150	0.3070	2148.9	19.99	49.05	0.3782	325.72	345.70	381.70	190	3.001	21,009	21.30	477.0	1.591	3184.4	3427.5	3473.1
151	.3186	2230.3	20.03	50.91	.3834	338.05	358.91	395.15	191	3.419	23,934	21.34	543.4	1.779	3627.7	3906.0	3951.8
152	.3304	2312.8	20.06	52.84	.3887	350.55	372.33	408.81	192	3.955	27,687	21.37	628.6	2.020	4196.6	4520.1	4566.2
153	.3436	2404.9	20.09	54.88	.3946	364.51	387.28	424.00	193	4.672	32,704	21.40	742.2	2.342	4957.0	5341.0	5387.4
154	.3569	2498.4	20.12	57.00	.4006	378.68	402.49	439.45	194	5.674	39,715	21.43	901.3	2.793	6019.6	6488.2	6534.8
155	0.3711	2597.4	20.16	59.26	0.4070	393.69	418.59	455.79	195	7.172	50,207	21.47	1139.2	3.468	7609.9	8205.2	8252.0
156	.3858	2700.6	20.19	61.61	.4136	409.33	435.41	472.85	196	9.663	67,643	21.50	1534.7	4.589	10253	10959	11006
157	.4012	2808.1	20.22	64.06	.4205	425.62	452.90	490.58	197	14.62	102,333	21.53	2321.0	6.819	15511	16734	16782
158	.4176	2922.9	20.25	66.67	.4279	443.03	471.59	509.51	198	29.25	204,715	21.57	4642.4	13.40	31029	33488	33536
159	.4347	3043.0	20.29	69.40	.4356	461.23	491.14	529.30	199	111.40	779,870	21.60	17682	50.37	118206	127620	127668
									200			21.63					

Low Rent Housing—

[*Concluded from page* 496]

should show this when the steam flow varies less than 20 per cent (10 per cent above and 10 per cent below the average demand) in an interval of 20 min.

The heating system must be tight enough so that a 15 in. vacuum produced on a cold system will not be reduced to less than 10 in. within 60 min after the pump has been stopped.

The heating system must circulate noiselessly and each radiator must heat throughout to the trap with a pressure differential range of 2 in. to 6 in. of mercury between the steam supply and the return.

Furthermore the control system must be capable of preventing overheating within the range of temperature of 75 to 68 F.

Zone control valves should be located so that each valve will have about the same steam demand. The variation should not exceed 10 per cent.

Risers: When exposed in rooms, should be considered as heating surface.

Supply and Return Mains: Should pitch down and be free of pockets, sags or lifts throughout their entire run.

Complete suggested specifications may be had upon application to the United States Housing Authority, Washington, D. C.

[*The appendix on fuel consumption formulas is to be published next month.*]

Gaskets and Closures—

[*Concluded from page* 501]

profile type bellows gaskets. Referring to Fig. 2, the ring type gasket, held in circular grooves, consists of symmetrical halves in contact in a plane at right angles to the pipe axis and united by peripheral welding. Radial grooves cut across the contact surfaces admit fluid of steam pressure into the bellows, the pressure being partially taken up by the outer wall of the bellows and partially added to the bolt pressure exerted upon the gasket surfaces in the usual manner.

If P is the unit fluid pressure, then the force F due to pressure within the bellows (see Fig. 2) is: $F = P\,(D^2 - d^2)\,\pi/4$. It is obvious that the bellows forms a pocket and that proper arrangements have to be made for drainage.

[*Other types are to be described in a second article.*]

Heating-Piping-Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

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COUNCIL ANNOUNCES OCTOBER MEETING

The Council has authorized a Fall Meeting of the Society and preparations are under way to hold a two day session in Atlanta October 30 and 31 under the auspices of the Atlanta Chapter.

The need of more technical meetings has been recognized and the success of this first Fall Meeting will indicate whether a permanent schedule of three meetings a year will be adopted.

A meeting in Atlanta opens new horizons and will bring the influence of the ASHVE to a greater number of engineers interested in its work.

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Heating Low-Cost Homes

By Robert K. Thulman,* Washington, D. C., and L. E. Seeley,** New Haven, Conn. (*MEMBERS*)

IN a paper, The Scientific Approach to the Housing Problem, by Vannevar Bush, formerly dean of engineering at M. I. T. and now president of Carnegie Institute, Dr. Bush says, "In our enormous country with its wide range of climate, methods of construction, and habits of the population, no single striking solution (of housing) is evident or should reasonably be expected." While this statement is particularly pertinent to the heating problem, it should not mean that solutions to the problem could be made without regard to some rather fundamental considerations. It is the object of this paper in part to present some features upon which a substantial agreement should be reached. These general questions, if resolved, can serve as guideposts which should lead to commendable contributions by those interested in the several phases of heating.

The scope of this paper is limited to a discussion of the heating of low-cost homes. In this field the home owner is least protected by adequate specifications and architectural and engineering advice. In this range the operative-builder finds it advantageous to avail himself of every opportunity to save money, frequently at the expense of quality.

The potential number of low-cost homes yet to be built warrants considerable thought in every phase, especially, it is believed, in the question of heating. The opportunity for constructive development and profitable business should be a vital consideration of every person or manufacturer engaged even remotely in the manufacture and design of products which enter into residential heating. Thus there exists a situation wherein a great field for new business and new development is offered and wherein there is a comparatively small amount of protection for the consumer.

By low-cost homes is meant, for the purposes of this paper, those homes intended to be occupied by families with an income of less than $2000 a year. These homes may be owned or rented. They may cost $1000 or $5000. In other words, the first cost is not the criterion, but the success of the undertaking will ultimately be judged by the ability of the income group for which the housing was designed to continue to occupy that housing. The simple fact is that out of the yearly income, among other things, the cost of shelter, including the cost of heat, must be afforded.

What general questions ought to be raised with a view to discovering a common viewpoint to the heating problem? The following are suggested:

(*a*) What standards of comfort should be secured?

(*b*) Should it be an invariable practice to provide for some minimum quantity of domestic hot water per day?

(*c*) To what extent are structural matters definitely related to heating?

(*d*) What is the proper basis for determining standards for the cost of heating?

Answers must necessarily be suggested if discussion is to be fruitful.

First of all, as a people, human beings live in permanent and generally substantial buildings. Clothing and diet are substantially uniform. Response to thermal environment runs in a predictable manner. The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has established standards of comfort for human beings in enclosed spaces. These findings, while different from those in England, are generally accepted as correct here. Assuming these standards are conducive to both comfort and health, there seems to be no alternative but to agree that low-cost homes, like any others, shall be heated in accordance with the accepted standards as found in the HEATING, VENTILATING, AIR CONDITIONING GUIDE of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS dealing with temperature, uniformity, air motion, humidity, etc. It follows that heating systems shall be capable of maintaining these standards in a home in any weather likely to occur during the heating season and that their capacity shall meet the design load as determined by methods of heat loss calculations as found in THE GUIDE. It is recognized that the heating systems in many existing homes fail to secure this standard of comfort, but that does not make a good argument against this proposed standard. It in fact gives the heating industry a greater opportunity for service than ever before.

Secondly, it is commonly known that in general public health is improved by increased use of water services. If there is an acknowledged value in the use of ample quantities of domestic hot water, it is a service which does not depend upon climate or geography. The cost of hot water will be more nearly a uniform cost per family than that of heating. It is suggested that a low-cost home should have from 30 to 50 gal of domestic hot water available per day. While this domestic hot water need not be supplied by the heating system as an additional service, there will be instances when the heating system will supply this water and in these cases the performance of the heating plant must be considered.

Third, climate varies so much in different sections of the country that large differences in heating costs seem inevitable. The number of degree days may vary from about 1000 to 10,000. The variation in climate, therefore, will influence the maximum capacity of a heating system over a wide range varying about 400 per cent for a given structure. This range can be reduced by insulation, control of window sizes and generally better construction to about 100 per cent. In other words, an acceptable low-cost house can be built with maximum heat losses varying between 25,000 and 50,000 Btu per hour.

It is recognized that some structural changes may be made, due to considerations of heat loss, which do not increase the cost of construction. On the other hand,

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Table 1—Budget Amounts for Heat and 30 Gal of Hot Water Per Day for Various Annual Incomes

Degree-Day	$1000	$1250	$1500	$1750	$2000
1,000	48.00	55.63	63.50	71.63	80.00
2,000	50.00	58.13	66.50	75.13	84.00
3,000	52.00	60.63	69.50	78.63	88.00
4,000	54.00	63.13	72.50	82.13	92.00
5,000	56.00	65.63	75.50	85.63	96.00
6,000	58.00	68.13	78.50	89.13	100.00
7,000	60.00	70.63	81.50	92.63	104.00
8,000	62.00	73.13	84.50	96.13	108.00
9,000	64.00	75.63	87.50	99.63	112.00
10,000	66.00	78.13	90.50	103.13	116.00

Table 2—Budget Amounts for Heat and 50 Gal of Hot Water Per Day for Various Annual Incomes

Degree-Day	$1000	$1250	$1500	$1750	$2000
1,000	56.00	63.63	71.50	79.63	88.00
2,000	58.00	66.13	74.50	83.13	92.00
3,000	60.00	68.63	77.50	86.63	96.00
4,000	62.00	71.13	80.50	90.13	100.00
5,000	64.00	73.63	83.50	93.63	104.00
6,000	66.00	76.13	86.50	97.13	108.00
7,000	68.00	78.63	89.50	100.63	112.00
8,000	70.00	81.13	92.50	104.13	116.00
9,000	72.00	83.63	95.50	107.63	120.00
10,000	76.00	86.13	98.50	111.13	124.00

where a question of heating systems for a given home arises where one may contemplate the use of insulation and the other not, it seems clear that insulation is a part of the cost of heating and should be included. If a climate is so severe as to demand insulation, then the cost should be a part of the cost of the heating system.

Again it may be that the chimney cost could vary depending upon the heating equipment chosen and the kind of fuel burned. The chimney might be a relatively inexpensive metal or composition *vent* or a standard tile-lined brick chimney. Here again if the cost of the chimney is added to the cost of the heating system because of the characteristics of that system, a truer picture of costs is obtained. It is suggested then as concerns part (*c*) that some recognition be given to the fact that certain structural features may properly become a part of the heating system cost.

Fourth, it is an accepted fact that the cost of heating is the largest single operating cost in a home. It is especially important where small incomes must buy so many necessities that costs not only be as low as possible but that they be reasonably predictable in advance. It is particularly urged that heating systems for low-cost homes should not be analyzed in terms of first or initial cost, but in terms of operating and maintenance costs. It meets a situation more realistically to express it in terms related to yearly income. If the operating cost of a heating system can not be afforded, that system should not be installed. Another reason in favor of stressing operating and maintenance cost is that this amount over the life of the heating equipment will be several times greater than the first cost of the system itself. In contrast to this, construction with high resistance to heat transmission would probably last as long as the building and require no maintenance. In that case the yearly cost would bear a direct relation to first cost. First cost should then be the sole criterion by which equivalent construction should be judged. It is suggested that the yearly budget for heating and domestic hot water should not exceed the values given in Tables 1 and 2.

A yearly operating budget requires that the performance of heating devices be known. This requires that products be tested in a manner that reveals pertinent economic data. A number of test codes have been devised from time to time and many are quite satisfactory in directing what measurements shall be made and in what manner. A common difficulty is that these codes may not always be followed. Again where test data are obtained they often are not published. Without test data, the economic performance of various combinations of heating devices leaves too much as a matter of sheer conjecture.

Even the problem of determining the capacity of devices so that a heating system will be adequate, to say nothing of economical, is not satisfactory. There are rating codes available in some cases, but often the only indication of capacity is a catalog rating which may or may not be founded upon a well-known and generally approved basis. It is suggested that a rating code is necessary for all classes of equipment and particularly that the code should indicate values that will be pertinent to the actual behavior and economy of the heating system as actually installed and used.

There is a lack of such codes or standards at the present time, though codes are now being prepared or revised by various groups. The practical danger is that progress of the several groups will not be uniform or that the character of the codes may not be of a uniform quality. This means that a group which presents its product in a way better suited to a determination of its economic value than some other groups may enjoy a competitive advantage not entirely justified by the product itself.

The Federal Housing Administration is authorized by the National Housing Act and its purposes are, first, to encourage building construction, and, second, to encourage improvements in housing standards. Recent amendments to the Act have stimulated construction of houses in the low-cost range by making financing of these houses attractive. As a result of the activities of the Federal Housing Administration and largely because of the recent amendments to the Act, better than 50 per cent of newly constructed houses costing under $5000 have been financed with insured mortgages. As a result of the large number of houses which have been submitted for mortgage insurance, it is felt that the experiences of the Federal Housing Administration on the question of heating should be of interest and value relative to the questions raised in the forepart of this paper. But before discussing these experiences in detail, a brief explanation of the procedure of the Federal Housing Administration might be in order. The Administration is required to determine, prior to a commitment to insure, that the property with respect to which the mortgage is issued is economically sound. The heating system, a vital part of the property, must, therefore, meet this test. In addition the buildings and the heating system must conform with standards prescribed by the Administration and these standards, by way of improving housing standards, must give consideration to health and comfort.

The amount of the mortgage on any particular property is determined by weighing the various factors which determine its valuation as distinguished from its cost. This system includes rating of the *mortgage pattern*, the borrower, the location, and the property itself.

In relation to the rating of the property, the mechanical equipment must be rated. The rating of this feature reflects the degree of adequacy, durability, and operating economy to perform the functions for which it is designed, considering the number and type of people likely to occupy property of this class. This is not done by maintaining a list of approved equipment, devices, or systems. Consequently, the heating equipment or system is considered for each home on its own merits. This procedure means that a great variety of combinations of heating equipment may be submitted to the FHA. Since this is so, it is obvious that more than casual knowledge of the many heating devices must be available. It is often true that catalog information itself is not enough and must be augmented by much more specific information to enable proper consideration of a particular kind of equipment with reference to its adequacy, durability, and operating economy. The establishment of standards by the Administration for other features of the house, such as space, window area; beam sizes, and plumbing, has been facilitated by reason of the existence of codes, specifications, and requirements which have been developed in the past and in which provision is made for constant revision as conditions require.

Because such codes lend themselves to the aims or objectives of the FHA, they can be incorporated directly in the requirements of the Administration with satisfactory results. For example, there are three basic codes for plumbing, and a comparison of them indicates that they do not differ in any essential features which affect performance or adequacy. Direct reference is made by the FHA to the Minimum Requirements for Plumbing of the Department of Commerce as an FHA requirement. Due to lack of heating codes of this character, there is little if any protection against the use of inadequate and poorly designed heating systems, the operating cost of which will be far in excess of the low-cost home owner's ability to pay. All considerations relating to safety are taken care of by direct reference to the Requirements of the National Board of Fire Underwriters. These requirements are based on minimum standards for safety from fire and explosion and are not concerned with economy of operation except insofar as safe equipment, because of inherent durability, may contribute toward lower maintenance and depreciation costs.

Concerned as it is with a large amount of building construction, it is natural that many complaints will arise. Experience has shown that a major proportion of these complaints in certain localities are on heating. In one territory it was found that a classification of the complaints on heating showed them to be due largely to excessive operating cost. This is one example of what could be avoided if proper and adequate technical data had been available at the time the commitments to insure the mortgages on these particular houses were made. In order to correct a condition of this kind, the FHA has resorted to the promulgation of specific rulings designed to deal with an existing situation in a particular territory, or has requested the industry, whose products may appear to be most directly responsible for the condition, to prepare performance specifications to correct future troubles or to revise existing codes and specifications which do not quite cover the situation, but which could be made to do so by amendment or revision. In order to avoid the liability involved in accepting inadequate or uneconomic heating systems and when the industry concerned is slow to prepare necessary corrective specifications, the Federal Housing Administration can avail itself of its prerogative to write down those properties submitted which include systems which experience has shown to be of poor quality. The amount of the mortgage can be modified up or down depending upon the quality and performance characteristics of the heating system.

For example, if the equipment is wisely chosen and installed, it may enhance the value of the property to the full extent of its cost. If, however, the cost of the equipment is too great in relation to the cost of the structure or if the typical buyer cannot afford the cost of operation of the equipment, it will not enhance the value of the property to the full extent of its cost. Conditions of this kind must necessarily be reflected in the face value of the mortgage. Obviously, the former methods, namely the use of the ruling or request to the industry for protective standards, are more expedient and usually more effective in achieving the objectives of adequacy, durability, and operating economy.

Recently in one of the Federal Housing Administration's offices a large number of complaints was received on heating of small homes. It was found that in every instance an oil burner was used. The oil burner industry was requested to prepare a specification to correct the trouble. As a result of industry meetings, a program was developed which included the establishment of an installation specification, evidence of compliance with which was incorporated in the form of a certificate to be posted in the basement. This constituted the first part of the program and was incorporated in a ruling made effective in the insuring office most concerned. Each installation was required to be tested and the results of the test noted on the certificate.

The industry continued its program covering standards for the manufacture of the burner itself and, through the coordinating facilities of the National Bureau of Standards, has incorporated the entire scheme in a Commercial Standard. One of the most interesting characteristics of this procedure is that the installation and performance codes so definitely assure improved operating economy. This characteristic is common to a greater or lesser degree in all standards established under this procedure inasmuch as consumer representation is included in their development. When a commercial standard exists, the Federal Housing Administration is in a position to include it as a part of its requirements. Lacking facilities of its own for testing, the Federal Housing Administration makes use of the National Bureau of Standards for determination of compliance with specifications and also uses the experience and knowledge of the Bureau in an advisory way.

It should be of interest to include in this paper an account of the variety of heating systems which have appeared in the field of low-cost homes. Practically every known system of heating has been utilized and there is reason to believe that each of these has some place. This variety has included everything from the most simple form of circulating heater to the fully automatic central heating plant, and includes all kinds of fuels and fuel burning devices. There has been a certain geographical significance with respect to the type of heating system used, but this seems to be gradually

disappearing. As a result there has been a tendency to use equipment found acceptable in a warm climate in localities where there may be some doubt of its adequacy, mainly because of the tendency to seek a cheaper system without due regard for operating economy or adequacy of performance. This trend as it develops may introduce difficulties due to performance and should make it evident that codes or standards are increasingly important.

There appears to be a trend toward the development of individual units which can be installed regardless of whether or not there is a basement, and one feature of this sort of development is the tendency to confine the equipment in a very small space. Since space is available vertically in a small house to a greater extent than horizontally, the procedure is to build the component parts of the unit one on top of the other. While there has been a tendency toward the elimination of the basement, this has been somewhat retarded by the complications it presents in the question of storage space including storage of fuel; also, it is affected by construction requirements in the colder climates which specify that footings shall be carried down below frost line; and, finally, because of the requirements of the basementless house that the heating medium be circulated mechanically with accessories, which add to the operating cost.

There are certain types of heating systems which have been characteristic of a particular locality. For example, the warm air system seems to be prevalent in the territory bordering on the Great Lakes and extending out into the Middle West. This may be explained by the wide public acceptance of that form of heating. This does not mean that other forms of heating which may be suitable for that particular territory could not be introduced. Hot water heating systems found in the houses of certain cities along the Eastern Seaboard are largely used because of public acceptance, and this, too, is no reason why other forms of heating could not be successfully used there. Another example of the influence of public acceptance is found through the belt extending from North Carolina westward to Arkansas. In this territory almost all houses are provided with a fireplace. As a result, the improved circulating type of fireplace finds a ready acceptance in this teritory. However, it has also been noted that where automatic equipment has been installed in homes already equipped with a fireplace, there has been a tendency to use the fireplace to a much lesser extent. While it is true that there are many homes in the extreme southern parts of the country which are not ordinarily equipped with any separate heating device, it is a fact that there is no state of the forty-eight which has not recorded at some time or another a temperature of zero degrees or lower, and there is only one weather station in the whole United States which has never recorded a temperature below freezing. This should indicate that, while there may not be actual suffering in these southern homes, there is at least discomfort at times. While many of these homes do not require complete heating systems, they still should have some form of heating device. In Southern California it has been common practice to build houses of an *open* type of construction, eliminating the building paper and permitting a comparatively large amount of infiltration. Consequently, in Southern California the unvented gas heater has been used to a large extent. With the use of insulating materials to provide greater summer comfort and generally tighter construction, the unvented heater presents a serious problem due to condensation. Therefore, such heaters should be separately vented or other means of heating introduced.

There is a definite trend toward the decrease of the design loss of the house by tighter construction, use of insulating materials, and more compact plans, which directly affects the maximum capacity of heating systems and particularly automatic firing devices. As a result of this trend, there is a lack of small capacity devices available. At the present time, this trend is resulting in an increase in the use of small gas-fired equipment where low capacities are not a serious problem. If a continuousness of heat output is to be obtained as preferred to highly intermittent heat, it would seem that the attention of the heating industry should be directed toward the development of devices with a capacity much smaller than those in present use.

While there is a tendency toward the development of automatic devices, this does not mean that development of devices to burn solid fuel manually should not continue, because automatic control is necessarily associated with higher cost and is necessarily limited in its application to a relatively small percentage of the possible occupants of low-cost homes. The availability of wood, particularly in rural localities, suggests the desirability of developing wood-burning equipment which possesses definite qualities of convenience and economy. It is axiomatic that lower costs create a greater market.

So far as future developments for heating the low-cost homes are concerned, there will probably be a continued improvement in all types of heating systems burning all types of fuels. These improvements will be made with the necessary considerations for reduced size, better performance, and greater operating economy. There is reason to believe that certain social considerations growing out of reduced space and the need for privacy may result in an increased use of the combination of a heating unit with a definite distribution system so that isolated rooms may be properly heated.

A reduction in heat losses without a corresponding reduction in hot water demand has developed the use of domestic hot water as the heating medium. Although not new, this development will probably continue. While there will be further attempts to reduce heat loss, it should be pointed out that too great a reduction in infiltration loss may be made at the expense of ventilation. The question might ultimately arise as to whether or not special provisions for ventilation may not have to be made.

Conclusions

1. Agreement on certain general questions seems necessary to the basic approach to the problem of heating low-cost homes.
2. The field experience of the Federal Housing Administration reveals that the heating system is one of its most serious concerns.
3. Solution of the problem must contemplate the availability of pertinent performance data and standards.
4. All recognized methods of heating and their associated devices participate in the heating of low-cost homes.
5. Future development of certain devices will require that they be made for heating loads smaller than at present.

Study of Summer Cooling in the Research Residence for the Summer of 1938

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating and Air Conditioning Association* and the University of Illinois.

DURING the summer of 1937 an investigation[1] was undertaken to determine the temperature and humidity conditions that could be maintained in the Research Residence with a mechanical condensing unit having a capacity approximately 40 per cent less than the 30,000 Btu per hour used during the summers of 1932 to 1936 inclusive.

heating system has been described in previous papers.[1, 2] In every respect the state of the Residence during 1938 was the same as that during the preceding summer. The arrangement of the forced-air duct system and fan was also the same for the two summers and was such that the cooled air could be delivered entirely through the registers on the first story, entirely through the registers on

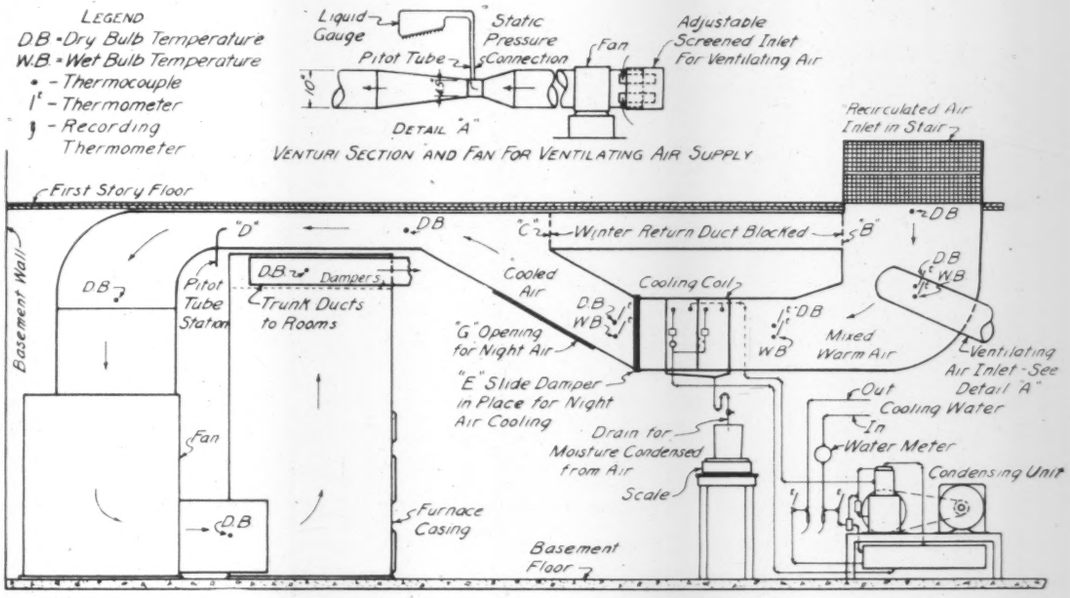

Fig. 1—Diagram of cooling plant with mechanical refrigeration

These tests with a small capacity, mechanical condensing unit were extended in 1938 and, in addition, were supplemented by tests made with the unit operated at a capacity of 30,000 Btu per hour. In the latter case, cooled air was delivered only to the rooms on the second story.

Description of Cooling Equipment

The Research Residence together with the forced-air heating system has been described in previous papers.[1, 2] In every respect the state of the Residence during 1938 was the same as that during the preceding summer. The arrangement of the forced-air duct system and fan was also the same for the two summers and was such that the cooled air could be delivered entirely through the registers on the first story, entirely through the registers on the second story, or through the registers on both stories.

The arrangement of the cooling plant, shown in Fig. 1, was practically the same as that fully described in a previous publication[1] except that two evaporator units were installed, and provision was made to use either one unit alone or both units in parallel. The maximum capacity of the condensing unit was approximately 30,000 Btu per hour with both evaporator units in use and the compressor running at 429 rpm. However, for the first part of the season (series 1-38, 2-38) the capacity of the condensing unit was reduced to 18,000 Btu per hour by changing a pulley on the motor, which lowered the compressor speed, and by closing a valve in the refrigerant line, which cut off one of the two evaporators.

A study of heat flow through the exposed west and north walls of the northwest bedroom of the Research

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[1]ASHVE Research Paper—Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock, and E. L. Broderick. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May and June 1938, pp. 337-344 and 421-427.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

[2]Investigation of Summer Cooling in the Warm-Air Heating Research Residence, by A. P. Kratz, M. K. Fahnestock, and S. Konzo. (University of Illinois Engineering Experiment Station Bulletin 290, January, 1937.)

Residence was begun in 1937.[3] These wall sections consisted of lap siding, building paper, sheathing, studding space, wood lath and plaster with rough sand finish. This study was extended in 1938 to include tests on an insulated wall section. For this purpose the west wall studding space, which was 5⅝ in. deep, was filled with mineral wool. The calculated coefficients of heat transmission for the uninsulated and insulated walls were respectively, 0.25 and 0.042 Btu per square foot per hour per degree Fahrenheit at a wind velocity of 15 mph. The north wall was not insulated. The thermocouples and the heat flow meters which were used in 1937 were also used in 1938 tests.

[3]Loc. Cit. See Note 1.

Table 1—Fan Data

(Calculations based on air density of 0.075 lb per cubic foot)

Series No.	Capacity of Unit Btu per Hour	Air Distribution	Air Volume, Cfm	Air Weight, Lb per Hour	Air Changes[a] per Hour
Cooling with Outdoor Air at Night					
1-38 and 2-38	.....	Both Stories	1560	.....	6.6
4-38	.....	Both Stories	1890	.....	8.0
Air Recirculation During Day Including Ventilating Air					
1-38	18,000	Both Stories	1020[b]	4590	4.3
2-38	18,000	2nd Story only	880[b]	3960	7.4[c]
3-38 and 4-38	30,000	2nd Story only	1040[b]	4680	8.7[c]
Ventilating Air					
For all series	.....	.....	228	1024	1.0

[a]Outdoor air admitted for ventilation amounting to one air change per hour included in these values.
[b]Quantity of air through wet coil.
[c]Based on second story only.

Method of Conducting Tests

Continuous records were made during the summer of air temperatures in the various parts of the house and in the different sections of the duct system. During the periods of operation, observations were made of the weight of water entering and leaving both the condensing unit and the house, the head and suction pressures, and the electrical inputs to the compressor and fan motors.

During all of the tests the windows in the attic remained open and those on the first story remained closed. The recirculated air was returned through the return air grille located in the stair risers in the first story hall. For the purpose of cooling with outdoor air at night, the attic door and 11 windows on the second story were opened in the evening when the effective temperature outdoors became equal to the effective temperature indoors. The dampers were set and the basement door opened, so that the fan delivered outdoor air

Table 2—Complete Data and Results for Tests—1938

Date 1938	Test No.	Series No.	Weather Data: Degree-Hours Above 80 F	Degree-Hours Above 85 F	Total Sunshine in Hours and Minutes	Outdoor Air: Maximum Temp. During Test, F	Minimum Temp. Previous to Test, F	Avg During Period Requiring Cooling: Temp., F	Hum., Per Cent	Indoor Air Conditions: At End of Night Cooling: Temp., F	Hum., Per Cent	Avg During Period Requiring Cooling: Temp., F	Hum., Per Cent	Temperature Difference: Outdoor Max. to Avg Indoor, F	Avg Outdoor to Avg Indoor, F	Night Air Cooling Previous to Test Period: Duration, Hours	Power, Kwhr
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
6-21	1	1-38	39.7	3.0	15:06	87	62	82.7	39.6	71.8	60.9	76.8	51.3	10.2	5.9	11.25	5.2
6-22	2	1-38	20.9	0	7:10	85	64	79.4	52.3	72.7	63.7	77.0	59.4	8.0	2.4	10.67	5.1
6-23	3	1-38	75.8	29.3	12:51	92	64	86.9	51.4	73.7	71.9	78.4	60.6	13.6	8.5	10.50	5.0
6-24	4	1-38	7.4	0	3:10	82	72	77.9	82.5	77.6	80.9	77.2	66.4	4.8	0.7	9.83	4.6
6-25	5	1-38	52.3	20.0	9:35	90	69	85.9	59.2	75.2	83.5	78.1	66.7	11.9	7.8	8.33	3.8
7-1	6	1-38	22.0	2.0	4:43	90	70	82.4	73.3	75.8	82.8	77.2	72.5	12.8	5.2	0	0
7-2	7	1-38	1.0	0	2:25	80	68	73.3	79.8	74.9	85.7	76.6	76.2	3.4	—3.3	12.92	6.3
7-3	8	1-38	5.0	0	9:40	82	69	80.3	62.4	73.8	80.4	77.1	67.1	4.9	3.2	17.25	0
7-4	9	1-38	3.0	0	12:35	82	63	79.0	54.7	72.0	61.9	77.0	58.3	5.0	2.0	11.67	5.6
7-5	10	1-38	13.4	0	13:34	82	63	80.3	49.8	71.5	65.3	77.1	54.9	4.9	3.2	11.67	5.4
7-6	11	1-38	51.2	11.4	14:49	87	65	82.9	59.0	72.4	66.7	77.5	56.3	9.5	5.4	11.50	5.6
7-7	12	1-38	71.7	29.6	12:25	90	68	87.2	54.5	75.3	71.4	78.7	61.0	11.3	8.5	9.17	4.4
7-8	13	1-38	45.1	6.9	10:43	86	68	83.2	61.0	75.3	72.9	77.8	61.1	8.2	5.4	12.50	6.3
7-9	14	1-38	60.4	13.8	14:56	90	66	85.3	49.8	75.3	70.2	77.8	57.0	12.2	7.5	9.25	4.5
7-10	15	1-38	83.2	32.2	12:39	90	68	85.8	53.7	75.1	68.1	78.5	57.2	11.5	7.3	10.75	5.1
7-11	16	1-38	93.7	38.4	11:57	96	74	85.6	56.3	77.6	72.2	79.4	57.7	16.6	6.2	8.42	4.0
7-12	17	1-38	79.6	34.6	10:50	92	70	85.5	57.8	76.9	70.5	79.5	61.3	12.5	6.0	9.25	4.3
7-13	18	1-38	38.4	2.6	6:20	86	70	79.7	67.3	75.8	65.1	77.4	59.6	8.6	2.3	10.50	5.1
7-14	19	1-38	38.6	0	14:41	85	68	81.6	45.8	75.4	57.5	77.3	51.1	7.7	4.3	0	0
7-15	20	1-38	9.1	0	12:52	84	62	80.4	50.2	72.1	62.1	77.5	53.8	7.5	2.9	12.00	5.9
7-16	21	1-38	52.1	12.4	11:55	88	64	83.4	54.9	71.8	66.1	77.7	56.9	10.3	5.7	12.25	6.1
7-19	22	1-38	23.1	0	14:35	86	64	82.1	51.0	71.2	66.9	77.1	55.5	8.9	5.0	11.92	5.8
7-20	23	1-38	32.5	7.4	10:02	87	66	83.9	57.4	72.6	68.5	77.2	62.1	9.8	6.7	11.75	5.6
7-23	24	1-38	16.3	0	14:50	85	62	80.6	50.9	71.5	74.8	77.3	59.4	7.7	3.3	13.00	6.3
7-24	25	1-38	35.0	0.7	14:36	88	62	82.1	59.3	70.9	68.5	77.3	59.8	10.7	4.8	12.17	5.8
7-25	26	1-38	70.5	24.2	12:51	91	62	85.2	59.2	73.1	68.5	78.2	61.0	12.8	7.0	10.92	5.1
7-26	27	1-38	27.9	6.6	8:15	89	69	82.4	61.7	75.2	72.1	77.4	63.8	11.6	5.0	9.00	4.2
7-27	28	2-38	13.4	0.3	5:25	87	68	79.2	76.7	73.8	83.1	77.2	74.0	9.8	2.0	16.00	7.6
7-28	29	2-38	3.9	0	13:08	87	67	82.9	56.9	73.6	82.5	77.5	62.2	9.5	5.4	13.00	6.0
7-29	30	2-38	10.0	0	12:04	84	65	80.8	53.4	73.3	73.1	77.6	58.9	6.4	3.2	11.25	5.3
7-30	31	2-38	20.6	0	12:46	83	64	81.0	69.0	72.5	70.4	77.4	64.7	5.6	3.6	12.00	5.8
7-31	32	2-38	2.6	0	6:45	81	69	80.1	69.9	75.3	80.0	76.5	65.9	4.5	3.6	11.33	5.4
8-1	33	2-38	1.0	0	8:20	85	64	79.4	71.4	72.4	75.3	76.4	70.1	8.6	3.0	12.50	6.0
8-2	34	2-38	48.9	10.4	14:18	88	67	84.4	64.9	73.6	81.4	77.0	64.9	11.0	7.4	11.50	5.4
8-15	43	4-38	24.3	0	3:15	83	71	81.8	73.1	No Night	Cooling	78.0	63.4	5.0	3.8	0	0
8-16	44	4-38	19.1	0	4:05	84	70	81.1	72.7	75.4	84.8	78.4	76.5	5.6	2.7	9.50	6.0
8-20	45	4-38	49.0	11.8	7:55	89	66	81.7	68.3	72.0	72.9	78.0	71.0	11.0	3.7	12.42	7.4
8-23	46	4-38	77.4	34.0	11:17	95	71	86.4	63.8	75.2	79.8	77.8	62.2	17.2	8.6	12.00	7.9

through the duct system to both stories until 7:00 a. m. The windows and doors were closed at 7:00 a. m. and the dampers were changed so that the fan started delivering recirculated air and the outdoor air admitted for ventilation. The fan was allowed to run continuously during the day and night. The quantity of air delivered by the fan for each test series and for cooling with outdoor air at night is shown in Table 1.

The Residence was operated strictly on the schedules described in connection with the following discussion of the different test series:

Simultaneous Cooling on Both Stories (Series 1-38)

After 7:00 a. m. the air was recirculated on both stories. When the average temperature on the second story rose to about 77 F, the condensing unit, operating at a capacity of 18,000 Btu per hour, was started and cooled air was distributed to the rooms on both stories.

The method of operation used in this series was the same as that for Series 1 of 1937 except that the cooling plant was started at a lower indoor temperature. In extremely mild weather the cooling plant operated intermittently and maintained an indoor temperature of 77 F. However, the heat absorbing capacity of the unit was so small that on most days when cooling was required, a slight rise in indoor temperature occurred as the test progressed, and the plant operated continuously during the whole period.

Cooling on Second Story with Small Capacity Unit (Series 2-38)

The air was distributed only to the second story rooms after 7:00 a. m. When the average temperature on the second story rose to about 77 F, the condensing unit, operating at a capacity of 18,000 Btu per hour, was started and cooled air was distributed only to the rooms on the second story.

Cooling on Second Story to Determine Rate of Reduction of Temperature on Second Story (Series 3-38)

The air was distributed only to the first story rooms after 7:00 a. m. and the condensing unit, operating at a capacity of 30,000 Btu per hour, operated intermittently maintaining a dry-bulb temperature of approximately 79 F. At some time during the early evening, when the temperature of the air on the second story had risen to a value exceeding 82 F, the dampers were changed so that all of the cooled air was distributed to the second story rooms alone. The cooling plant operated continuously until the average temperature on the second story was reduced to 78.5 F.

Cooling on Second Story with Large Capacity Unit (Series 4-38)

The method of operation was the same as that for Series 2-38 except that the condensing unit was operated at a capacity of 30,000 Btu per hour. The unit was con-

Table 2—Complete Data and Results for Tests—1938

Operation of the Fan				Overall Test Period			Compressor Operation					Total Moisture Condensed, Lb	Heat Absorbed by Cooling Coil During Overall Test Period, Btu			Ratio Moisture to Total Load
Recirculation Previous to Test Period		Recirculation During Test Period		Start of Cooling with Condensing Unit to Start of Night Air Cooling			Time in Hours	Power Input, Kwhr	Condenser Water				Load Due to Moisture	Sensible	Total	
									Quantity in Gallons	Temperature						
Duration, Hours	Power, Kwhr	Duration, Hours	Power, Kwhr	Start	End	Duration, Hours				Inlet, F	Rise, F					
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35
7.33	3.0	6.00	2.7	2:20p	8:20p	6.00	5.70	7.4	293.6	58.3	40.4	12.65	14,250	84,550	98,800	0.144
8.25	3.6	5.25	2.3	3:15p	8:30p	5.25	2.72	3.5	158.0	58.2	37.4	9.73	11,050	38,450	49,500	0.223
5.08	2.0	9.08	4.2	12:05p	9:10p	9.08	8.92	11.4	532.0	58.1	38.5	48.92	55,100	115,400	170,500	0.323
0	0	15.67	7.1	7:00a	10:40p	15.67	14.00	18.2	821.0	58.3	39.2	102.59	114,650	153,350	268,000	0.428
2.25	1.0	9.00	4.3	9:15a	6:15p	9.00	9.00	11.7	519.0	58.5	40.5	64.43	71,800	103,200	175,000	0.410
4.33	1.8	2.75	1.4	3:20p	6:05p	2.75	2.58	3.6	150.5	58.7	40.4	19.79	22,890	27,710	50,500	0.454
3.00	1.2	3.75	1.8	10:00a	1:45p	3.75	1.05	1.3	63.3	58.9	40.5	8.84	9,840	11,490	21,330	0.461
5.75	2.5	6.58	3.1	12:45p	7:20p	6.58	4.58	5.9	259.0	59.4	39.8	27.09	30,200	55,700	85,900	0.352
9.00	4.1	3.33	1.5	4:00p	7:20p	3.33	2.65	3.7	142.3	59.6	39.8	9.16	11,280	25,920	47,200	0.239
8.50	3.9	4.00	1.6	3:30p	7:30p	4.00	3.82	4.9	199.6	59.1	40.2	10.90	12,510	54,390	66,900	0.187
6.25	2.9	8.58	3.8	1:15p	9:50p	8.58	8.50	11.1	462.9	59.1	40.3	40.39	45,500	109,900	155,400	0.293
2.50	1.2	9.00	4.1	9:30a	6:30p	9.00	9.00	11.7	510.2	59.0	40.4	52.20	58,950	112,750	171,700	0.343
2.50	1.0	12.25	5.7	9:30a	9:45p	12.25	12.08	15.7	676.6	59.0	40.3	70.44	82,600	144,559	227,159	0.363
2.25	1.0	11.00	5.2	9:15a	8:15p	11.00	11.00	14.3	597.5	59.1	40.2	47.75	53,800	146,275	200,075	0.269
3.00	1.5	12.58	6.0	10:00a	10:35p	12.58	12.33	15.9	683.0	59.4	40.0	55.57	62,750	164,720	227,470	0.276
0	0	14.75	7.0	7:00a	9:45p	14.75	14.50	18.8	818.2	59.0	40.4	76.89	88,250	187,096	275,346	0.320
0.58	0.3	12.92	6.1	7:35a	8:30p	12.92	12.67	16.6	730.0	59.3	39.8	78.24	88,900	153,200	242,100	0.367
3.33	1.5	20.67	9.8	10:20a	7:00a	20.67	18.03	23.1	1014.0	59.8	39.4	102.00	114,200	218,700	332,900	0.313
1.50	0.5	10.50	4.9	8:30a	7:00p	10.50	9.85	12.7	516.0	59.4	39.6	25.88	29,900	140,100	170,000	0.176
9.25	4.0	2.50	1.3	4:15p	6:45p	2.50	2.25	2.8	117.4	60.0	39.2	5.61	6,580	31,750	38,330	0.172
7.00	3.0	6.25	2.9	2:00p	8:15p	6.25	6.00	7.5	334.4	58.9	40.3	24.87	28,300	83,580	111,880	0.253
8.00	3.5	4.25	2.0	3:00p	7:15p	4.25	4.00	5.3	212.2	59.9	39.3	14.28	16,190	53,260	69,450	0.233
5.25	2.3	3.50	1.7	12:15p	3:45p	3.50	3.18	4.3	176.1	59.6	39.5	15.46	17,220	40,780	58,000	0.297
9.33	4.1	2.67	1.2	4:20p	7:00p	2.67	2.42	3.2	130.4	60.0	39.2	8.06	9,110	33,470	42,580	0.214
8.50	3.8	4.42	2.1	3:40p	8:05p	4.42	4.20	5.4	234.1	60.1	39.0	21.20	23,700	51,950	75,650	0.313
4.67	2.1	10.33	4.8	1T:40a	10:00p	10.33	10.33	13.6	586.0	59.4	39.7	59.85	68,200	125,400	193,600	0.352
2.05	0.9	5.95	2.6	9:03a	3:00p	5.95	5.78	7.2	326.0	59.1	39.9	35.72	40,720	67,580	108,300	0.376
4.67	2.1	6.33	2.9	11:40a	6:00p	6.33	4.17	5.6	244.7	60.2	38.9	32.07	36,080	43,220	79,300	0.455
5.83	2.6	6.92	3.0	12:50p	7:45p	6.92	6.50	8.6	371.0	60.0	39.2	35.13	40,100	81,000	121,100	0.331
7.00	3.0	5.00	2.2	2:00p	7:00p	5.00	4.50	5.8	240.9	59.7	39.5	16.50	19,300	59,920	79,220	0.244
7.00	3.1	5.67	2.5	2:00p	7:40p	5.67	5.17	6.7	290.5	59.9	39.2	32.10	36,200	58,700	94,900	0.382
8.00	3.5	3.50	1.6	3:00p	6:30p	3.50	3.50	4.4	204.8	61.0	38.2	23.63	26,800	38,400	65,200	0.411
10.17	4.5	2.33	1.1	5:10p	7:30p	2.33	1.42	1.9	82.8	60.8	38.4	8.85	10,050	16,450	26,500	0.379
5.15	2.7	8.85	3.6	12:09p	9:00p	8.85	8.85	11.7	520.0	60.4	38.8	60.68	68,600	99,300	167,900	0.409
5.00	2.9	9.50	5.9	12:00	9:30p	9.50	5.18	12.0	590.0	59.7	38.2	59.08	65,400	122,100	187,500	0.349
7.22	4.1	8.28	4.9	2:13p	10:30p	8.28	2.02	4.8	239.5	.....	.....	16.40	.....	.....	.....	.....
9.43	5.7	3.07	1.8	4:51p	7:55p	3.07	0.75	2.0	83.9	.....	.....	2.49	.....	.....	.....	.....
5.42	3.1	9.35	5.7	12:25p	9:46p	9.35	6.08	15.0	694.0	59.3	37.9	57.86	64,050	155,050	219,100	0.292

trolled to maintain a temperature of approximately 78 F on the second story.

Results of Tests

Simultaneous Cooling on Both Stories (Series 1-38)

The results obtained from this series of tests were practically the same as those obtained from the work done during the preceding summer. General data and results from all of the tests are shown in Table 2.

The operating characteristics of the cooling plant can best be illustrated by the results obtained on a typical day. For this purpose the test made on June 23, 1938, was selected as typical of the operation obtained when the rooms on both stories were cooled (Series 1-38), and the results are shown in the first column of Table 3 and in Fig. 2. On this test the second story windows had been open during the preceding night from 8:30 p. m. to 7:00 a. m., and outdoor air had been circulated by means of the basement fan. The windows were closed at 7:00 a. m. and recirculation of the air in the house was begun. The condensing unit was started at 12:05 p. m.

The tests made during the summer of 1937 had shown that with high humidity conditions at the start of a test, a starting temperature of about 78 F resulted in an indoor effective temperature higher than the 75 deg representing the upper border line of comfort. In 1938, however, the indoor effective temperature at the start of the tests was not allowed to exceed 75 deg. This was accomplished by starting the condensing unit at the lower indoor temperature of about 77 F, as shown in Fig. 2. The condensing unit operated continuously from 12:05 p. m. to 9:00 p. m. maintaining the dry-bulb temperature between 77.4 F and 79.2 F. Three hours after the condensing unit started the relative humidity stabilized at about 59 per cent and the effective temperature did not exceed 75 deg. The indoor conditions during this period could be considered as entirely satisfactory.

It is evident from the curve of indoor dry-bulb temperature in Fig. 2 that the condensing unit was insufficient in capacity to maintain a constant temperature. If the temperature of the outdoor air had been materially higher, the rise in the indoor temperature would have been greater and the maximum indoor effective temperature would probably have attained a value above that represented by the upper limit of the comfort zone. Since the season was comparatively cool, no test data under severe weather conditions were obtainable.

In the study made in 1937 a relationship was shown between the maximum indoor air temperature obtained in the house and the daily median outdoor temperature, or the average of the daily maximum and minimum outdoor temperatures. The results obtained have been reproduced and are shown as broken-line curves in Fig. 3. The maximum temperatures that were obtained in the house, when cooling was accomplished only by the use of outdoor air at night, are shown as the upper curve. For a day on which the median temperature did not exceed 75.5 F the indoor maximum temperature was 79 F and conditions were reasonably satisfactory if the relative humidity was not too high. The lower broken-line curve, which gives the maximum indoor air temperatures obtained when the small capacity condensing unit was operated, indicates that on days when the daily median temperature was 83 F or less the maximum indoor air temperature did not exceed 80 F.

The results obtained from tests made during 1938, shown as circles in Fig. 3, confirmed the results obtained in 1937, except that the starting of the condensing unit at a lower indoor temperature resulted in a lower maximum temperature on mild days. The lower starting temperature also required that the cooling plant be operated on days when cooling with outdoor air at night alone would have given fairly satisfactory indoor condi-

Table 3—Typical Operating Data and Results

1. Date	6-23-38	8-2-38	8-23-38
2. Test No.	3	34	46
3. Series	1-38	2-38	3-38
4. Time of Day	2:10 p.m.	2:20 p.m.	4 p.m.
5. Outdoor Air			
Dry-Bulb, F	90.5	87.0	90.9
Wet-Bulb, F	73.7	75.2	77.7
Dew-Point, F	66.5	70.7	72.9
Relative Humidity, per cent	44.3	57.5	54.5
Specific Humidity, grains per pound dry air	97.5	112.7	121.9
6. Indoor Air, average breathing level temperature			
1st Story, F	77.7	76.5	77.6
2nd Story, F	77.8	76.8	77.8
7. Indoor Air, average at breathing level			
Dry-Bulb, F	77.8	76.7	77.7
Wet-Bulb, F	68.6	68.6	69.1
Dew-Point, F	64.2	64.7	64.9
Relative Humidity, per cent	62.2	66.0	57.6
Specific Humidity, grains per pound dry air	89.8	91.4	82.3
8. Indoor Air, entering return grille			
Dry-Bulb, F	77.4	76.7	77.1
Wet-Bulb, F	67.9	68.4	66.4
Dew-Point, F	63.2	64.5	60.9
Relative Humidity, per cent	61.0	65.6	56.5
Specific Humidity, grains per pound dry air	86.6	90.6	79.6
9. Ventilating Air			
Dry-Bulb, F	88.1	85.1	88.1
Wet-Bulb, F	72.4	74.8	76.0
Dew-Point, F	65.5	70.7	71.3
Relative Humidity, per cent	46.1	61.2	56.5
Specific Humidity, grains per pound dry air	94.2	113.1	115.5
10. Mixed Air, entering cooling coil			
Dry-Bulb, F	79.5	78.5	79.4
Wet-Bulb, F	68.9	69.8	68.6
Dew-Point, F	63.9	65.8	63.3
Relative Humidity, per cent	58.0	64.3	57.0
Specific Humidity, grains per pound dry air	88.7	95.1	86.9
11. Mixed Air, leaving cooling coil			
Dry-Bulb, F	67.6	66.6	58.8
Wet-Bulb, F	63.1	63.4	57.3
Dew-Point, F	60.6	61.7	56.3
Relative Humidity, per cent	78.0	83.9	91.8
Specific Humidity, grains per pound dry air	78.9	81.9	67.6
12. Air Temperature drop through cooling coil, F	12.0	11.9	20.6
13. Temperature cooled air leaving registers, average, F	71.0	70.6	68.8
14. Air Temperature rise in ducts and casing, F	3.5	4.0	10.0
15. Basement Air Temperature at breathing level, F	72.8	73.1	73.7
16. Cooling Coil			
Gross Face Area, sq ft	2.07	2.07	2.07
Net Free Area, sq ft	1.02	1.02	1.02
17. Air Velocity through coil			
Gross Face, fpm	493	425	563
Free Area, fpm	1,000	863	1,020
18. Moisture condensed from air, pounds per hour	5.88	7.04	10.59
19. Heat given up by air; total Btu per hour	19,720	19,250	34,900
Heat due to moisture in air			
Btu per hour	6,620	7,950	11,720
Per cent of total heat	33.6	41.3	33.6
Sensible heat			
Btu per hour	13,100	11,300	23,180
Per cent of total heat	66.4	58.7	66.4
20. Water Temperature in condenser			
Inlet, F	58.2	60.4	59.3
Outlet, F	95.0	99.2	97.9
Rise, F	36.8	38.8	38.6
21. Water Temperature in house			
Inlet, F	58.0	60.4	59.3
Outlet, F	95.2	98.8	97.8
Rise, F	37.2	38.4	38.5
22 Quantity of cooling water			
Pounds per hour	514	485	940
Gallons per hour	61.7	58.2	112.9
23. Heat absorbed by cooling water, Btu per hour			
Through condenser	18,920	18,820	36,280
Through house	19,120	18,625	36,190
24. Refrigerant pressures, gage			
Suction, pound per sq in	38	40	36
Discharge, pound per sq in	125	125	130
25. Nominal Rating of condensing unit with 140-lb head pressure, 55 F return refrigerant, 59 F condensing water and 74 F ambient air, Btu per hour	16,300	16,300	31,150
26. Compressor Speed, rpm	229	229	429
27. Compressor Motor Data			
Size, hp	2	2	2
Measured power rate, kw	1.28	1.32	2.47
28. Fan Speed, rpm	444	444	526
29. Fan Motor Data			
Size hp	0.75	0.75	0.75
Measured power rate, kw	0.43	0.44	0.60

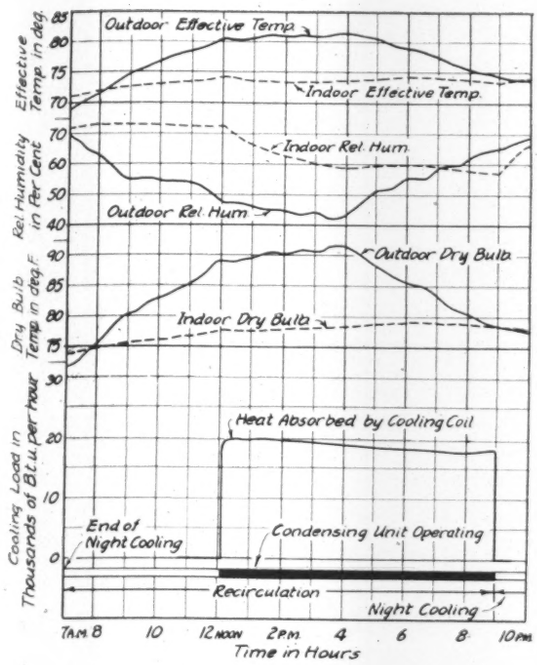


Fig. 2—Graphic log of air temperatures, relative humidities and cooling load. Test No. 2. Series 1-38. June 23, 1938. Capacity 18,000 Btu per hour

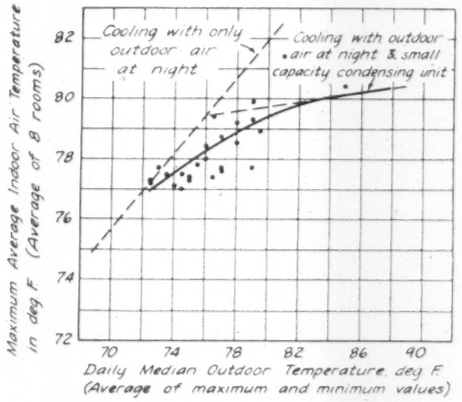

Cooling with only outdoor air at night
Cooling with outdoor air at night & small capacity condensing unit
Maximum Average Indoor Air Temperature in deg. F. (Average of 8 rooms)
Daily Median Outdoor Temperature, deg. F. (Average of maximum and minimum values)

Fig. 3—Variation of daily maximum indoor temperature with outdoor median temperature. Research Residence data. Summer of 1938. Series 1-38

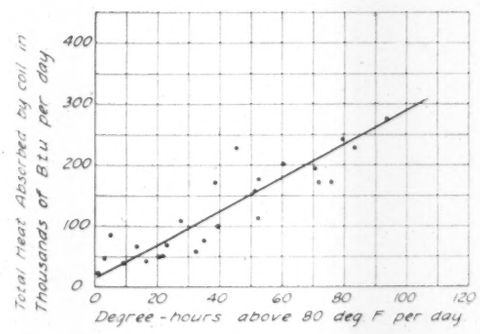


Fig. 4—Heat absorbed by cooling coil per day. Research residence data. Series 1-38

tions. The probable values of the maximum indoor air temperature, for operating conditions as maintained in 1938, are shown by the solid line in Fig. 3 which coincides with the curve for 1937 in the upper range. Since the daily median temperature of 83 F was exceeded only on three days during the cooling season of 1938, the capacity of the condensing unit was sufficient to maintain satisfactory conditions for practically the entire season. However, the analysis presented in a previous paper[4] had indicated that for an average season the small capacity condensing unit would not have maintained satisfactory conditions on 10 days during the season. This number would undoubtedly be exceeded in a severe season.

The total cooling load for the day, expressed as the total Btu absorbed from the air in the total amount of time during which the machine was actually in operation is shown in Fig. 4, plotted against the degree-hours per day. The base value of 80 F was selected in this case rather than that of 85 F which has heretofore been used, because it was found that when the machine was started at an indoor temperature of 77 F it was operated on many days on which the maximum outdoor temperature was less than 85 F. Considerable deviations of the points from the mean curve are to be expected since wind and sun effects, which influence the cooling load, are not directly dependent upon the degree-hours. Also the amount of cooling with air from outdoors at night is indirectly reflected in the cooling load required for a given number of degree-hours the next day. The curve shown in Fig. 4 can be used to estimate the seasonal amount of heat absorption required under similar conditions of plant operation for any season for which the number of degree-hours above 80 F corresponding to each day in the season is known.

Cooling on Second Story with Small Capacity Unit (Series 2-38)

The test made on August 2, 1938, was selected as typical of the operation obtained when the small capacity condensing unit was used and cooled air was delivered only to the rooms on the second story. The results are shown in the second column of Table 3 and in Fig. 5. Outdoor air had been circulated through the entire house during the night preceding the test. At 7:00 a. m. the windows were closed, and recirculation was begun with all of the air distributed to second story rooms. The condensing unit was started when the temperature on the second story reached 77.5 F.

A rapid reduction in relative humidity occurred on the second story when the condensing unit was started at 12:09 p. m. This was accompanied by a small reduction in dry-bulb temperature and resulted in a marked decrease in effective temperature. The relative humidity and dry-bulb temperature on the first story were affected in a manner similar to those on the second story, but to a somewhat lesser degree. The result was that a satisfactory effective temperature was obtained on both stories with cool air introduced only on the second story. The return air grilles were located on the first story in the two lower stair risers, and this location undoubtedly accentuated the natural tendency for the cool air to gravitate to the lower story. In general, the results indicate that if partial cooling is to be used, it is advisable to locate the cool air outlets on the second story rather than

[4]Loc. Cit. See Note 1.

the first, particularly if the return air grille is located on the first story. It is possible that in some cases first story rooms remote from the stairway may not be greatly affected. This did not seem to be true in the Research Residence, in which case the whole lower story was affected by the cooling on the second story. It was observed, however, that without positive circulation of air on the first story, any additions to the usual number of occupants, or any additional heat sources, soon produced unsatisfactory conditions in the rooms more remote from the return grille and the stairway.

Cooling on Second Story with Large Capacity Unit (Series 4-38)

The results obtained when the condensing unit was operated at a capacity of 30,000 Btu per hour and cooled air was delivered only to the rooms on the second story are shown in the third column of Table 3 and in Fig. 6. A schedule of operation similar to Series 2-38 was used, with air distributed only to second story rooms after 7:00 a. m. following the cooling with outdoor air during the night. However, since the capacity of 30,000 Btu per hour was ample to maintain a constant temperature in the house, before the condensing unit was started the temperature on the second story was allowed to rise to 78.5 F instead of the 77.5 F which was used with the smaller capacity in Series 2-38. On account of the high humidity conditions prevailing a starting temperature of 78.5 F on the second story was used instead of approximately 80 F which was used in 1934. With this starting temperature the house was maintained at an average temperature of approximately 78 F.

It may be noted from Fig. 6 that even the comparatively low starting temperature of 78.5 F was accompanied by a very high humidity which resulted in an effective temperature higher than 75 deg at the start of the test. Also the intermittent operation of the cooling plant produced large fluctuations in relative humidity and the effective temperature rose above 75 deg during the off-periods at the beginning of the test. For most of the test period, however, the conditions were entirely satisfactory. Since effective temperatures above 75 deg occurred during the off-periods, it follows that conditions in the house were more comfortable on hot days, when the plant operated with long on-periods, than they were on mild days, when the plant operated with short on-periods and more frequent off-periods.

During the entire test the temperature on the first story approximated the temperature on the second story quite closely. Humidity conditions were even better since, as shown in Fig. 6, the relative humidity on the first story was approximately equal to the average of the extremes occurring on the second story, and so resulted in more uniformly satisfactory effective temperatures. The effectiveness of cooling on the first story was subject to the same limitations as those discussed in connection with Series 2-38.

Rate of Reduction of Temperature on Second Story (Series 3-38)

During the summer of 1937[5] a series of tests was made in which the first story was cooled during the day and the second story at night. The condensing unit operating at a capacity of 18,000 Btu per hour was used. When the cooled air was first distributed through the registers on the second story, the reduction in temperature and relative humidity was quite rapid. However, the reduction in temperature occurred very slowly after the first hour due to the effect of the heat absorbed in the structure, and amounted to 0.7 F per hour. The tests indi-

[5]Loc. Cit. See Note 1.

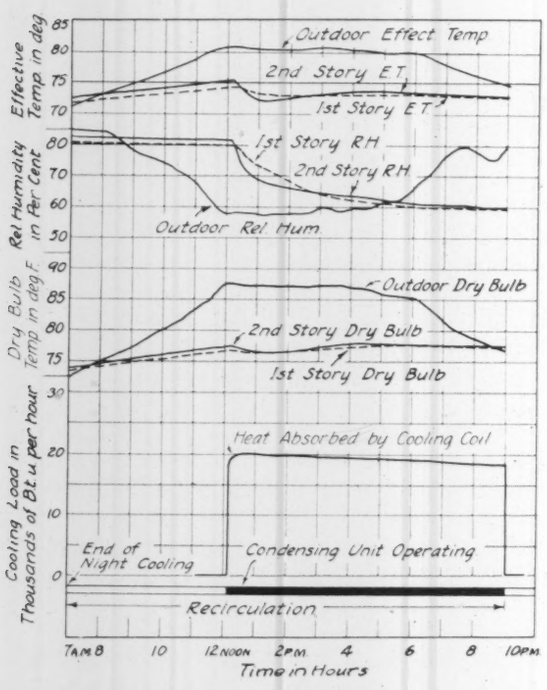

Effective Temp. in deg.
Outdoor Effect. Temp.
2nd Story E.T.
1st Story E.T.
Rel. Humidity in Per Cent
1st Story R.H.
2nd Story R.H.
Outdoor Rel. Hum.
Dry Bulb Temp. in deg. F.
Outdoor Dry Bulb
2nd Story Dry Bulb
1st Story Dry Bulb
Cooling Load in Thousands of Btu per hour
Heat Absorbed by Cooling Coil
End of Night Cooling
Condensing Unit Operating
Recirculation
7 A.M. 8 10 12 Noon 2 P.M. 4 6 8 10 P.M.
Time in Hours

Fig. 5—Graphic log of air temperatures, relative humidities and cooling load. Test No. 34. Series 2-38. August 2, 1938. Capacity 18,000 Btu per hour. Recirculation on second story only

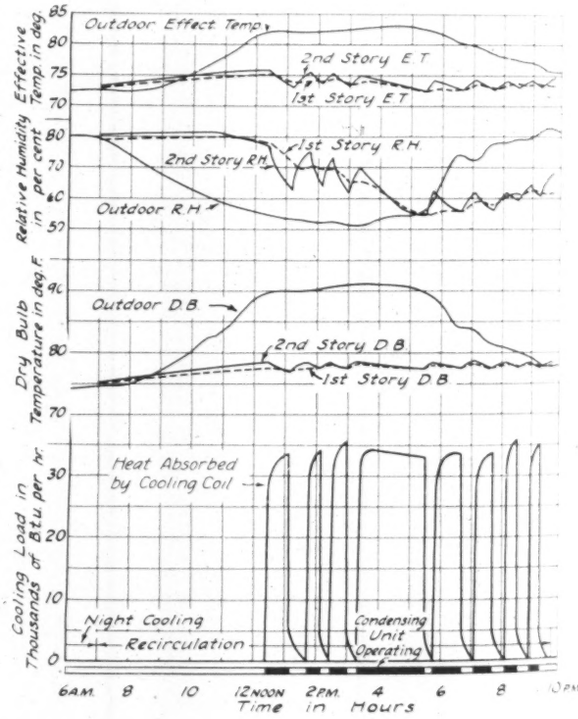


Fig. 6—Graphic log of air temperatures, relative humidities and cooling load. Test No. 46. Series 4-38. August 23, 1938. Capacity 30,000 Btu per hour. Recirculation on second story only

cated that a comparatively large plant would be required to effect a rapid decrease in indoor air temperature.

In order to obtain comparative data on the reduction of temperatures and relative humidities, a similar series of tests was made in 1938 with the condensing unit operated at a capacity of 30,000 Btu per hour. The operating characteristics for a typical day are shown in Fig. 7. From 2:00 p. m. to 6:30 p. m. cool air was delivered only to the rooms on the first story and the temperature was maintained at approximately 79 F. During this period the temperature on the second story rose to 83.3 F. From 6:30 p. m. to 9:40 p. m. cool air was distributed only to the rooms on the second story and the air temperature was reduced 4.8 F. During this same period the relative humidity on the second story was reduced from 66 per cent to 51 per cent and the effective temperature was reduced from 78.6 deg to 73.0 deg.

The curves in Fig. 7 showing the rate of cooling on the second story have been reproduced (curves No. 1) in Fig. 8, together with similar curves from two other tests. The curves labeled No. 2 show the results obtained when the temperature on the second story was allowed to rise to 87 F before cooling was started. The broken-line curves designated as No. 3 have been reproduced from the 1937 series of tests with the condensing unit operating at a capacity of 18,000 Btu per hour.

It may be observed from curves Nos. 1 and 2 that the rate of cooling on the second story as measured by the decrease in dry-bulb temperature, relative humidity, and effective temperature was practically the same for starting temperatures of 83.3 F and 87.0 F. However, this increase in the temperature at which cooling was started lengthened the time required to lower the temperature on the second story to 80 F, resulting in a total time of five and one-quarter hours instead of one and one-quarter hours. Assuming that it is desirable to cool the second story to 80 F in not more than three hours, the maximum allowable temperature at the beginning of the period should not exceed 85 F with a condensing unit operating at a capacity of 30,000 Btu per hour. For initial air temperatures exceeding 85 F a larger capacity plant would be required.

When the condensing unit was operated at a capacity of 18,000 Btu per hour (curves No. 3), the temperature on the second story was reduced 4.0 F in the first 3 hours, indicating that this capacity was sufficient when the starting temperature did not exceed 84 F. The relative humidities during this period were higher than those maintained when the large capacity plant was operating, which in turn was reflected in higher effective temperatures. In general, the curves in Fig. 8 indicate that an increase in the capacity of the condensing unit from 18,000 to 30,000 Btu per hour did not materially increase the rate of cooling on the second story during the first hour of operation, but did decrease the total time required to reach a predetermined temperature.

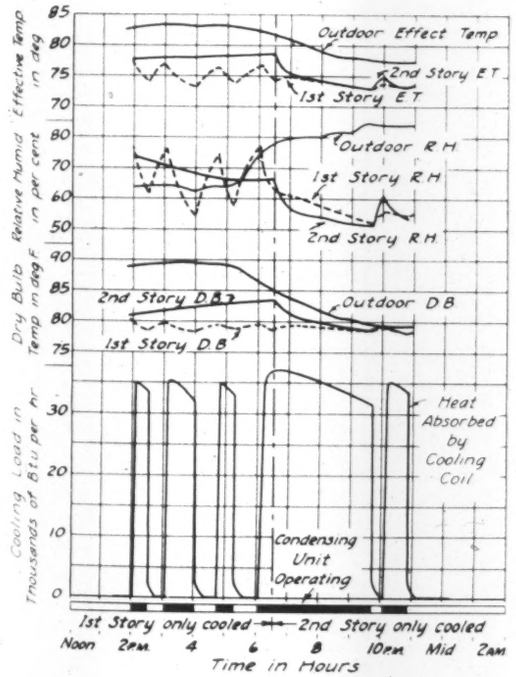

Outdoor Effect Temp
2nd Story ET
1st Story ET
Outdoor R H
1st Story R H
2nd Story R H
2nd Story D B
Outdoor D B
1st Story D B
Heat Absorbed by Cooling Coil
Condensing Unit Operating
1st Story only cooled
2nd Story only cooled
Noon 2pm 4 6 8 10pm Mid 2am
Time in Hours
Dry Bulb Temp in deg F
Relative Humid in per cent
Effective Temp in deg
Cooling Load in Thousands of Btu per hr

Fig. 7—Graphic log of air temperatures, relative humidities and cooling load. Test No. 40. Series 3a-38. August 9, 1938. Capacity 30,000 Btu per hour. All air to second story after 6:30 p. m.

Results Obtained with Heat-Flow Meters

The data obtained from the two heat-flow meter installations for a total of 20 days were plotted on a continuous chart which was used as a basis for the analysis of results. A typical section from the chart for the insulated section of the west wall, which was exposed to the sun in the afternoon, is shown in the lower portion of Fig. 9. From the 1937 data a similar day was selected and the temperature curves for the uninsulated west wall were plotted, as shown in the upper part of Fig. 9. It may be noted that the temperatures were continually changing and that thermal equilibrium was never attained.

The two intermediate curves in Fig. 9 show the actual amounts of heat passing through the inside surface of the uninsulated and insulated walls, as indicated by the readings of the heat-flow meter. The meter was located directly below the thermocouples installed to obtain the temperature gradients through the wall. The maximum values of the heat actually transmitted into the room, as

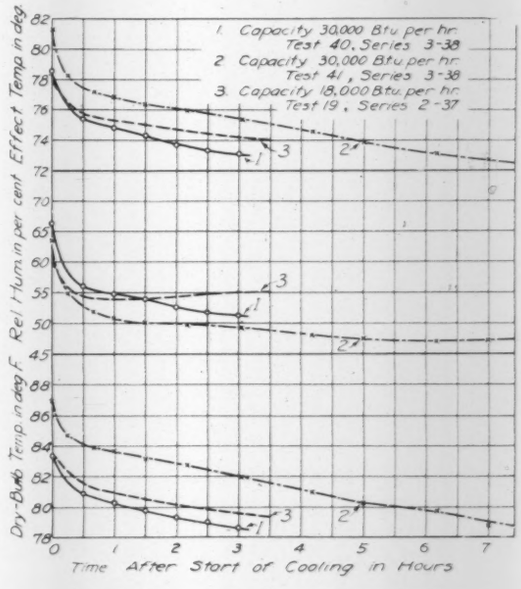

1. Capacity 30,000 Btu per hr. Test 40, Series 3-38
2. Capacity 30,000 Btu per hr. Test 41, Series 3-38
3. Capacity 18,000 Btu per hr. Test 19, Series 2-37
Dry-Bulb Temp in deg F
Rel Hum in per cent
Effect Temp in deg
Time After Start of Cooling in Hours

Fig. 8—Rate of cooling of second story

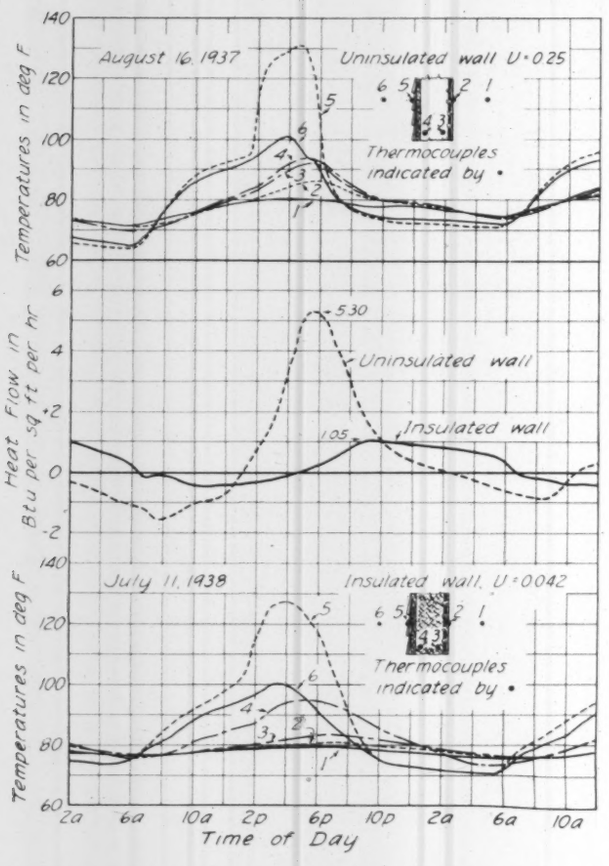


Fig. 9—Graphic log of temperatures and heat flow for uninsulated and insulated walls. Research Residence data for west wall. Seasons of 1937 and 1938

measured by the heat-flow meter, were 5.30 and 1.05 Btu per square foot per hour for the uninsulated and insulated walls respectively. The maximum rate of heat flow into the room for the uninsulated wall occurred at 5:30 p. m., whereas that for the insulated wall occurred approximately four hours later. The results obtained in 1937 indicated that the average time lag from maximum outdoor air temperature to maximum indoor air temperature was approximately two and one-half hours. The results from the 1938 tests indicate that this average time lag for the insulated wall was of the order of six and one-half hours. The total amount of heat per square foot transmitted into the room during a 24-hour period was obtained by integrating the area included between the curves of heat flow and the zero line. The values were 27.74 Btu per square foot for the uninsulated wall and 9.55 Btu per square foot for the insulated wall.

The curves shown in Fig. 10 represent average conditions based on data obtained over a wide range of weather conditions. Hence ratios based on these curves are more representative than similar ratios based on the individual days shown in Fig. 9. Both the data for the maximum rate of heat transmitted into the room and the total heat per square foot transmitted per day, as measured by the heat-flow meter, were plotted against the maximum value of the temperature of the outdoor surface of the wall. For a comparatively hot day in which the maximum surface temperature on the west wall was 130 F the ratio of the maximum rates of heat transmission for the uninsulated and insulated walls was approximately four to one, as compared with a ratio of 6.0 to 1 based on the calculated coefficients of heat transmission. The curves for total heat transmitted per square foot through the west wall per day, as shown in the lower part of Fig. 10, indicate that the ratio of the total heats transmitted through the uninsulated and insulated walls for a hot day was approximately 2 to 1. In both cases for a milder day, in which the maximum surface temperature was less than 130 F, the ratios were smaller.

From a practical standpoint the curves indicate that the application of a substantial amount of insulation reduces the maximum heat flow through the walls by an amount only two-thirds of that indicated by the ratio of the calculated coefficients of heat transmission for the walls. However, the use of insulation delays the time at which the heat input reaches a maximum value several hours beyond the time at which the design calculations of heat gain are generally made. Since the calculations of heat gain from a structure are based on maximum considerations, the fact that the maximum heat flow through an insulated wall occurs so much later than other maximum conditions indicates that some reconsideration of the methods employed in calculating heat gains in such structures may be advisable. In any event, the ratio of the maximum heat transmitted through the walls per square foot per hour has direct bearing on the relative sizes of plant required for structures with uninsulated walls and those with insulated walls. On the other hand, the ratio of total heat transmitted through the walls per square foot per day has direct bearing on the relative cost of operation, once the proper size of plants has been selected. The results indicate that the apparent reduction in the maximum rate of heat flow through walls alone, as calculated from the coefficients of heat transmission, would be five-sixths, or 83.3 per cent, while the actual reduction would be three-fourths, or 75.0 per cent. That is, the actual reduction would be 90.0 per cent of the apparent reduction. Hence, if the heat gain through walls was 25 per cent of the total heat gain, the actual reduction in heat loss, and therefore the permissible reduction

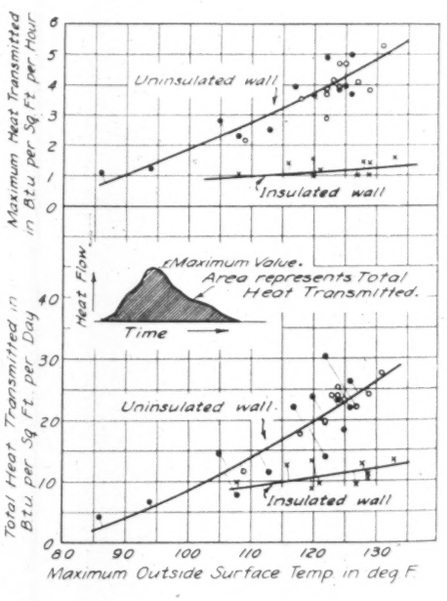

Maximum Heat Transmitted in Btu per Sq Ft per Hour
Uninsulated wall
Insulated wall
Heat Flow
Maximum Value. Area represents Total Heat Transmitted.
Time
Total Heat Transmitted in Btu per Sq Ft per Day
Uninsulated wall
Insulated wall
80 90 100 110 120 130
Maximum Outside Surface Temp in deg F

Fig. 10—Maximum and total heat transmitted per day through uninsulated and insulated walls. Research Residence data on west wall. Seasons 1937 and 1938

in plant size, to be effected by the use of insulation would be approximately 97 per cent of that calculated from the coefficients of heat transmission. In the case of the total heat transmitted daily through the walls the apparent reduction would again be five-sixths, or 83.3 per cent, but the actual reduction would be one-half, or 50 per cent, and the actual reduction would be 60 per cent of the apparent. In this case, if the heat gain through walls represented 25 per cent of the total, the actual reduction in daily heat gain, and hence the reduction in operating cost would be approximately 90 per cent of the apparent reduction as calculated from the coefficients of heat transmission. These results further indicate the necessity for additional investigation, probably under laboratory conditions, of the heat flow through walls as affected by cyclic variations in temperature on one or both sides.

in the wall was occurring simultaneously to the outdoors and to the indoors. The calculated values of heat transmission are of little significance under such conditions; the significant values being given by the heat-flow meter, which on account of its location on the inside surface of the wall measures the actual flow of heat into the room. It may also be observed from a study of the readings of thermocouples Nos. 3 and 4, that, following a day on which the maximum outdoor temperature reached 100 F, the temperatures in the studding space were definitely reduced to a temperature approximating that of the indoor air. Hence, apparently there was no material storage of heat in the insulation to be carried over to the following day and to become cumulative in the case of a succession of hot days.

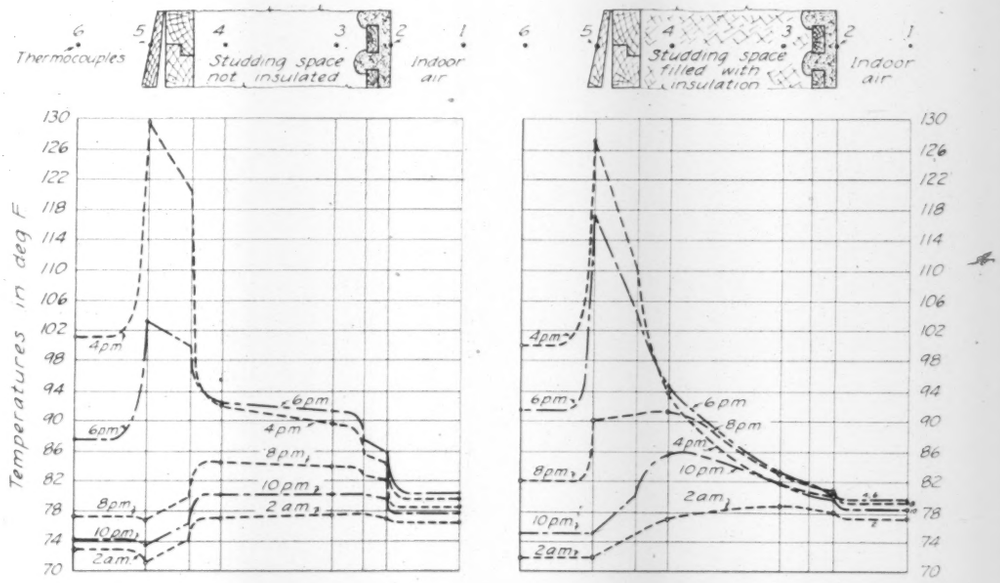


Fig. 11—Temperature gradients through uninsulated and insulated wall sections at different times of the day. Data from west wall. Seasons of 1937 and 1938

The comfort conditions maintained in a given space are dependent not only on the temperature and humidity of the air, but also on the surface temperatures of the walls, to which the occupants are exposed. It may be observed from the curves labeled No. 2 in Fig. 9 that the inside surface temperatures of the exposed wall varied from hour to hour. For the purpose of comparison, temperatures taken from Fig. 9 have been replotted and shown as temperature gradients through the walls in Fig. 11. These temperature gradients indicate that the inside surface temperature of the uninsulated wall attained a maximum value of 86 F, while that for the insulated wall attained a maximum value of only 81 F, as compared with an indoor air temperature of approximately 78 F in both cases. Hence, with the same indoor air temperature, the insulated wall provided conditions which were the more suitable for comfort.

The temperature gradients for both the uninsulated and insulated walls indicate that at certain times of the day the temperatures in the studding space were higher than those of both the indoor and outdoor surfaces of the wall. That is, under these conditions the heat flow

Summary and Conclusions

The following summary and conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

(1) In order to maintain the same effective temperature, the unit having the smaller capacity was operated to maintain an indoor dry-bulb temperature somewhat lower than that maintained with a unit having a larger capacity. Furthermore, it was advisable to allow for a rise in the indoor temperature on hot days when the cooling plant did not have sufficient capacity to maintain a constant temperature.

(2) The cooling capacity of 18,000 Btu per hour was found to be sufficient for both stories for days on which the maximum outdoor temperature rose to 95 F and the minimum outdoor temperature during the night preceding the test did not exceed 71 F. The daily median temperature (average of maximum and minimum outdoor temperature) of 83 F may be considered as the upper limit of the range of satisfactory application of the small-sized condensing unit.

(3) Subject to limitations over a small portion of the cooling season, the use of a condensing unit of small capacity was shown to be feasible for residences, especially in those cases in which

the initial cost of installation is a more important factor than the maintenance of ideal indoor conditions.

(4) In the case of the Research Residence, satisfactory cooling on both stories was accomplished by the introduction of cooled air on the second story only. The application of this principle in general may be limited by the arrangement of the first story, particularly with respect to the location of the return air grilles and the location of rooms remote from the stairway.

(5) By introducing air on the second story only and increasing the capacity of the condensing unit from 18,000 to 30,000 Btu per hour the time required for cooling the second story from 83.5 F to 80 F was reduced from approximately 2¼ hours to 1¼ hours.

(6) The studies made with heat-flow meters indicated that considerable discrepancy existed between the measured values and the calculated values of the heat flow through the wall. Furthermore, thermal equilibrium was never attained in the wall section under actual conditions.

(7) In the case of the uninsulated wall the average time lag between the attainment of maximum temperature outdoors and maximum temperature indoors was approximately 2½ hours. In the case of the insulated wall this time lag was approximately 6½ hours.

(8) The application of insulation to a frame wall materially reduced the maximum hourly heat flow and the total heat flow through the wall per day. However, the reduction in both the maximum and total heat flow was less than that indicated by the ratio of the calculated coefficients of heat transmission of the uninsulated and insulated walls.

(9) The application of insulation reduced the temperature of the inside wall surface thus resulting in conditions more conducive to comfort for a given air temperature than those produced by the uninsulated wall.

(10) The insulated wall cooled to a temperature approximating that of the indoor air by 2:00 a. m., thus indicating that there was no material storage of heat in the insulation to be carried over to the following day and to become cumulative in the case of a succession of hot days.

Acknowledgments

The results presented in this paper were obtained in connection with the summer cooling investigation (1938) in the Research Residence[6] at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to R. B. Engdahl, research graduate assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

[6]The Research Residence in Urbana, Ill., was built, furnished and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

First Fall Meeting in Atlanta

THE inaugural Fall Meeting of the Society will be held in Atlanta, Ga., October 30 and 31, with the Atlanta Chapter as hosts.

It is planned to have three Technical Sessions, a get-together luncheon, a banquet and dance and a program of special entertainment for ladies.

Since the July meeting, when the Council accepted the cordial invitation of the Atlanta Chapter members to be hosts for the Fall Meeting, a General Committee on Arrangements has been appointed consisting of: L. F. Kent, General Chairman; C. T. Baker, Vice-Chairman; C. L. Templin, Reception; S. W. Boyd, Registration and Sessions; E. W. Klein, Banquet; T. T. Tucker, Ladies; A. H. Koch, Sports; G. H. Brodnax, Jr., Transportation; M. M. Crout, Publicity; and C. B. Cole, Finance.

Broad St., Atlanta, looking north

According to the announcement of Pres. J. F. McIntire, the Council will hold its fall meeting at the Atlanta-Biltmore Hotel on October 29 preceding the general Society meeting.

The Atlanta Chapter members are enthusiastic about the prospect of entertaining the Society and as this was the first chapter organized in the south, its members keenly appreciate the responsibility of acting as hosts for the first Fall Meeting and the first Society meeting to be held in a southern city. True southern hospitality awaits ASHVE members in Atlanta October 30-31.

National Mineral Wool Day at World's Fair

The *National Mineral Wool Association* held a luncheon in the Casino of Nations at the World's Fair, New York on Thursday, July 13, in observance of National Mineral Wool Day. A preliminary gathering in the Casino of Nations at noon was followed by luncheon at 1:00 p.m. A tour of the grounds was conducted at 3:00 p.m. to 56 buildings, and later a number of those present visited the two summer comfort test houses at East Floral Park, L. I.

Guest speakers were Prof. F. B. Rowley, University of Minnesota, on Engineering Tests on Mineral Wool; T. G. Grace, New York State Director, Federal Housing Administration; R. H. Shreve, architect, on Mineral Wool as an Insulation for Masonry Walls; and A. W. Canney, on Rational Basis for Determining Thickness of Insulation.

The Effect of Vibration Upon Free Convection from Horizontal Cylinders

By R. C. Martinelli* and L. M. K. Boelter**

THE rate of heat transfer from a 0.75 in. diameter horizontal cylinder, being vibrated sinusoidally as a whole in a vertical direction, to surrounding water has been determined experimentally. The amplitude of vibration of the cylinder was varied from zero to 0.10 in. and the frequency of vibration from zero to 40 cycles per second. The difference in temperature between the cylinder and the surrounding fluid was varied from 8 F to 45 F.

The experimental results have been correlated using the dimensionless parameters, Nusselt (Nu), Grashof (Gr), Prandtl (Pr), and Reynolds (Re). In the Reynolds number used in the correlation the velocity term represents the root mean square velocity of the vibrating cylinder.

Velocities of vibration yielding magnitudes of Reynolds' Modulus less than a *critical value* (about 1000), have no effect on the rate of heat transfer from the cylinder within the range of experimental data, ($5 \times 10^5 \leq Gr \cdot Pr \leq 5 \times 10^6$). At these low velocities of vibration the heat loss from the cylinder is largely due to free convection forces; that is, the difference in temperature between the cylinder and the ambient fluid determines the rate of heat transfer.

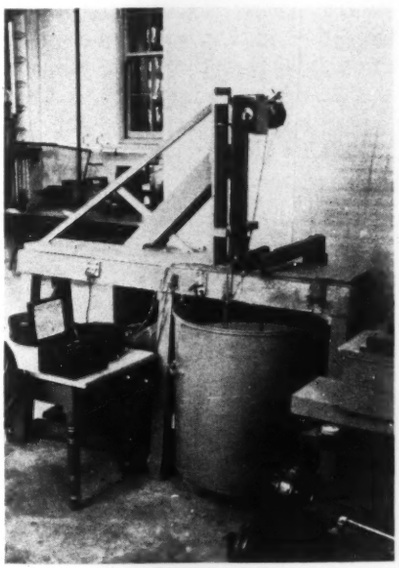


Fig. 1—Arrangement of apparatus

As Reynolds' Modulus (velocity of vibration) is increased beyond this critical value, the rate of heat transfer increases with an increased velocity of vibration. At a sufficiently high magnitude of Reynolds' Modulus, the effect of free convection becomes negligible and the forced convection due to the vibration controls.

Extrapolation of the data beyond the experimental limits for each variable, or their application to other boundary conditions, is not recommended.

*Engineer, Shell Oil Co., Martinez, Calif.
**Professor, Mechanical Engineering, University of California, Berkeley, Calif.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

Apparatus

The apparatus utilized to measure the rate of heat transfer consisted of a horizontal cylinder 0.75 in. in diameter and 12⅝ in. effective length which was vibrated as a whole in a vertical direction (normal to the cylinder axis), by means of a connecting rod and crank mechanism. An electric heater and thermocouples were enclosed in the cylinder.

The cylinder was placed in the center of a 36 × 36 in. cylindrical tank filled with water. A photograph of the experimental station is shown in Fig. 1, and views of the cylinder with connecting rod and vibrating mechanism are illustrated in Fig. 2.

The temperature of the cylinder, the temperature of the ambient water, the power input to the heater, the frequency of vibration and the amplitude of the vibration were measured during each run.[1]

The amplitude and frequency of vibration were kept constant for four runs, while the power input to the heater was varied in order to note the effect of the free convection modulus (Grashof's Modulus) on the rate of heat transfer.

Discussion of Results

The results of the tests are shown plotted in Figs. 3 and 4. Fig. 3 reveals the boundary modulus Nu (proportional to the film transfer factor from the tube to the water) as a function of the free convection modulus times the fluid property modulus ($Gr \cdot Pr$) for various magnitudes of Reynolds' Modulus (Re). It is noted that for no vibration and for small values of Re (less than 1000) the data obtained checks well with the free convection data for stationary tubes[2] obtained by others. As Reynolds' Modulus (Re) is increased beyond 1000, a new curve of Nu *vs* $Gr \cdot Pr$ is obtained for each magnitude of Re, the value of Nu increasing with increasing magnitudes of Re. In addition the variation of Nusselt's Modulus (Nu) with $Gr \cdot Pr$ (proportional to the free convection forces) decreases as the velocity of vibration increases until at a magnitude of Re equal to about 7000, Nu is independent of the free convection forces and depends upon Re only.

This result follows because the role of the free convection heat loss becomes of smaller importance as the vibrational velocity is increased.

Cross plots of the curves in Fig. 3 are shown in Fig. 4 and illustrate clearly the variation of Nu with Re

[1]Martinelli, R. C., and Boelter, L. M. K., The Effect of Vibration on Heat Transfer by Free Convection from a Horizontal Cylinder. (*Proceedings of the Fifth International Congress of Applied Mechanics*, Cambridge, Massachusetts, 1938, p. 578. John Wiley & Sons, 1939).

[2]McAdams, W. H., Heat Transmission. (McGraw Hill Co.)

for constant values of Gr · Pr. An algebraic expression relating Nu, Re and Gr · Pr has been derived semi-analytically:

$$\mathrm{Re} = \mathrm{Nu}\sqrt{12{,}000 - 20\frac{(\mathrm{Gr}\cdot\mathrm{Pr})^2}{\mathrm{Nu}^8}} \quad \ldots\ldots(1)$$

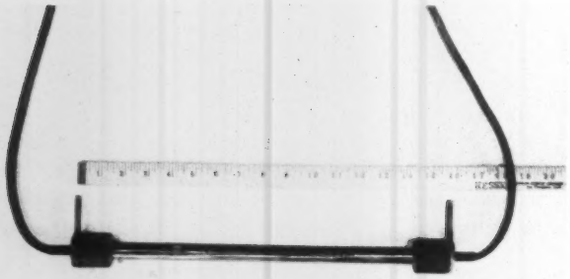
As seen in Fig. 4 the expression conforms to the experimental data quite well. Complete details of the derivation of equation (1) will be found in reference 1.

Several Examples of Application of Results

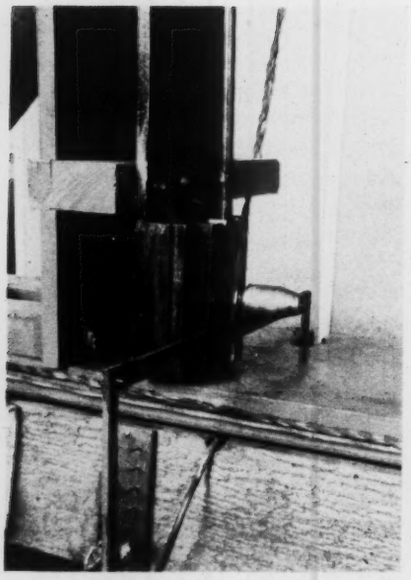
A tube ½ in. in diameter carrying steam is being used to heat water to 150 F. The steam temperature is 250 F. What amplitude of vibration (of the tube) is required to double the heat transfer rate if the frequency of vibration is 30 cycles per second?

1. Computation of the thermal conductance per unit area for stationary tubes. Reference 2, page 249, indicates $\mathrm{Nu} = 0.525\ (\mathrm{Gr}\cdot\mathrm{Pr})^{1/4}$, an empirical relation between Nu and Gr·Pr for stationary tubes applicable in the range $10^3 \leq \mathrm{Gr}\cdot\mathrm{Pr} \leq 10^7$. The product Gr·Pr may be rewritten as follows:

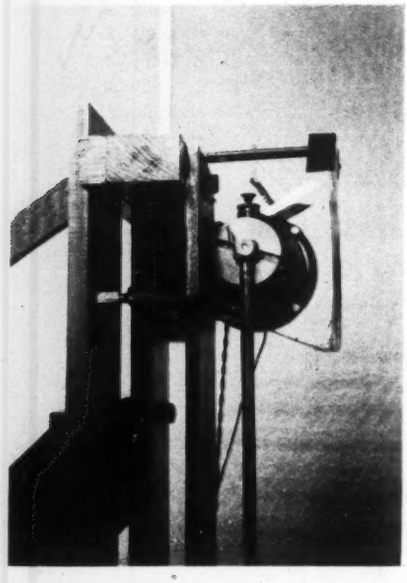
$$\mathrm{Gr}\cdot\mathrm{Pr} = \alpha d^3 \triangle t, \text{ where } \alpha = \frac{\rho^2 g^2 \beta C_p}{3600\,\mu k}\ (\mathrm{ft})^{-3}\ (\mathrm{F})^{-1}$$



a. Horizontal cylinder



b. Cylinder support



c. Connecting rod and crank

Fig. 2—View of cylinder with connecting rod and vibrating mechanism

Magnitudes of α for water and air are obtained from the literature:

$\alpha = 18.6 \times 10^8\ (\mathrm{ft})^{-3}\ (\mathrm{F})^{-1}$ at an average temperature of 200 F

$d = \frac{1}{2} \times 1/12 = 4.16 \times 10^{-2}$ ft

$\triangle t = 250\ \mathrm{F} - 150\ \mathrm{F} = 100\ \mathrm{F}$

$\therefore\ \mathrm{Gr}\cdot\mathrm{Pr} = 18.6 \times 10^8 \times (4.16)^{-2} \times 10^{-6} \times 100$

$= 13.4 \times 10^6$

Nu for stationary tubes

$\mathrm{Nu} = 0.525\ (13.4 \times 10^6)^{1/4} = 31.8.$

To double the film conductance per unit area Nusselt's Modulus (Nu) must be doubled. Thus, the magnitude of Nu required $= 31.8 \times 2 = 63.6$. Substituting the magnitudes of Nu and Gr·Pr into equation (1) yields

$$\mathrm{Re} = 63.5\sqrt{12{,}000 - 20\left[\frac{13.4 \times 10^6}{(63.6)^4}\right]^2}$$

$$= 6950 = \frac{a\,\omega\,d\,\rho}{\sqrt{2}\,\mu}$$

However:

$\omega = 30 \times 2\pi = 189$ radians per second

$d = \frac{1}{2} \times 1/12 = 4.16 \times 10^{-2}$ ft

$\mu/\rho = 0.34 \times 10^{-5}$ ft² per second for water at 200 F

$$\therefore\ a = \frac{1.414 \times 0.34 \times 10^{-5} \times 6950}{189 \times 4.16 \times 10^{-2}}$$

$= 4.23 \times 10^{-3}$ ft

$= 0.051$ in., the amplitude required to double the heat transfer rate at 30 cycles per second. The total displacement of the tube is twice this magnitude, that is, 0.102 in.

If a body is losing heat in air, the vibration is not as effective in increasing the rate of heat transfer as it would be if water is the ambient fluid. This fact may be shown by comparing the magnitudes of the velocities of vibration which yield the same value of Reynolds' Modulus (Re) in air and in water.

$$\text{Let } \left(\frac{a\,\omega\,d\,\rho}{\sqrt{2}\,\mu}\right)_{\text{air}} = \left(\frac{a\,\omega\,d\,\rho}{\sqrt{2}\,\mu}\right)_{\text{water}}.$$

For the same tube diameter,

$$d_{\text{air}} = d_{\text{water}};\ \text{then } \frac{(a\,\omega)\ \text{air}}{(a\,\omega)\ \text{water}} = \frac{(\rho/\mu)\ \text{water}}{(\rho/\mu)\ \text{air}}$$

$$(\rho/\mu)\,\text{water} = 2.93 \times 10^5\ \frac{\text{second}}{\text{ft}^2} \text{ at 200 F}$$

$$(\rho/\mu)\,\text{air} = 4.17 \times 10^3\ \frac{\text{second}}{\text{ft}^2} \text{ at 200 F}$$

The ratio of the vibrational velocity required in air to that required in water to give the same Re is 70.3:1. It follows that vibrating a body in air will have no effect on the rate of heat transfer, until it is vibrated about 70 times as fast as would be necessary in water.

The purpose of this paper has been to present quantitatively the effect of vibration on free convection. The effect of vibration in air at frequencies and displacement amplitudes normally encountered is negligible. However, in water the effect becomes appreciable, the increase over free convection heat transfer being four fold for Reynolds' Moduli equal to 10,000. For Reynolds' Modu-

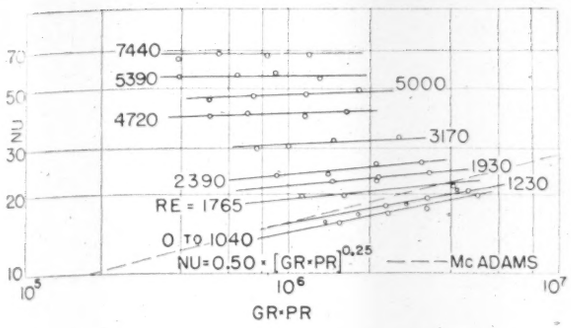


Fig. 3—Variation of the boundary modulus (Nu) with product of free convection modulus and fluid property modulus (Gr × Pr) for various magnitudes of Reynolds' Modulus (Re)

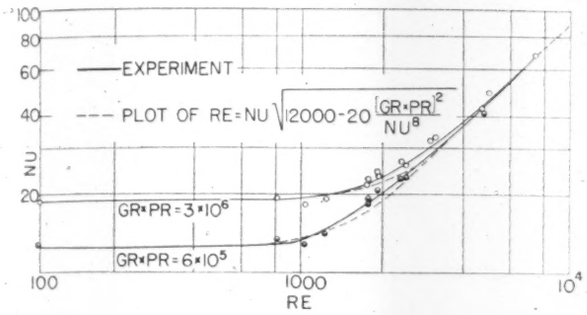


Fig. 4—Boundary modulus (Nu) as a function of Reynolds' Modulus (Re) for two magnitudes of (Gr × Pr). Experimental points and the graphical representation of the empirical equation

lus less than 1000 no appreciable effect of vibration is observed. The minimum magnitude of Reynolds' Modulus at which vibration becomes effective increases with an increase in free convection rate of heat transfer. For heat transfer to fluids similar to water, in apparatus subjected to vibration, a liberal design results if the apparatus behavior prediction is based upon data obtained from stationary equipment.

Acknowledgments

The authors wish to acknowledge their indebtedness to Wilson Tripp, who paved the way for this work; to C. J. Carlson, who worked with the authors at the beginning of the investigation; and to S. Polland and H. Johnson who obtained some of the experimental data. Technological assistance was had through WPA project No. 6090-5970.

Table of Nomenclature

a	= amplitude of displacement of the tube	feet
c_p	= specific heat of fluid	Btu per (pound) (F)
d	= diameter of tube	feet
f	= thermal conductance per unit area	Btu per (foot)^2 (hour) (F)
g	= gravitational acceleration = 32.2 ft per second^2	
k	= thermal conductivity of the fluid	Btu per (foot)^2 (hour) (F per foot)
t	= temperature	F
t_p	= temperature of pipe	F
t_w	= temperature of the ambient fluid	F
Δt	$= t_p - t_w$	F
α	$= \dfrac{\rho^2 g^2 \beta C_p}{3600\,\mu k}$ = property of fluid	feet^{-3} (F)$^{-1}$
β	= expansion coefficient of fluid	(F)$^{-1}$
γ	= weight density of fluid	pound per foot^3
μ	= viscosity of fluid	(pound) (second) per foot^2
ν	= kinematic viscosity of fluid	feet^2 per second
ω	= angular velocity of vibration	second^{-1}
$\omega/2\pi$	= frequency of vibration	second^{-1}
ρ	= mass density of fluid	pound second^2 per foot^4

All properties are evaluated at $(t_p + t_w)/2$.

Dimensionless Moduli

$\mathrm{Gr} = \dfrac{d^3\rho^2 g\beta\Delta t}{\mu^2}$ — Grashof's Modulus, the free convection modulus.

$\mathrm{Pr} = \dfrac{\mu c_p g}{3600\,k}$ — Prandtl's Modulus, the fluid property modulus.

St = Stanton's Modulus, another fluid property modulus.
Pr = 1/St.

$\mathrm{Re} = \dfrac{a\omega d\rho}{\sqrt{2}\,\mu}$ — Reynolds' Modulus, the forced flow modulus.

Exposition Advisory Committee Announced

The Sixth International Heating and Ventilating Exposition, to be held in Cleveland, Ohio, January 22-26, 1940, will be representative of all branches of the industry, judging by the list of companies who have already taken space. A total of 230 firms have engaged over two-thirds of the available exhibit space. The Exposition will be held in Lakeside Hall, a part of the Civic Center in Cleveland. This comparatively new building is conveniently situated and offers modern facilities adapted to exhibit requirements.

The Advisory Committee of the Sixth International Heating and Ventilating Exposition, composed of leaders in the industrial fields represented, has recently been announced. J. F. McIntire, Detroit, president of the ASHVE, is chairman of the committee and other ASHVE representatives are: F. E. Giesecke, College Station, Tex.; W. L. Fleisher, New York; E. H. Gurney, Toronto; M. F. Blankin, Philadelphia; J. H. Walker, Detroit; G. L. Tuve, Cleveland; John Paul Jones, Cleveland; L. T. Avery, Cleveland; and O. W. Motz, Cincinnati. Representatives of other organizations on the Advisory Committee are: W. B. Henderson, *Air Conditioning Manufacturers Association;* Gardner Poole, *American Society of Refrigerating Engineers;* Ole Singstad, *American Institute of Consulting Engineers;* R. M. Conner, *American Gas Association;* F. H. Dewey, *Oil Burner Institute;* L. R. Taylor, *National Warm Air Heating and Air Conditioning Association;* W. J. Snow, *Cleveland Engineering Society;* W. C. Murray, *Institute of Boiler and Radiator Manufacturers;* J. M. Frank, *Industrial Unit Heater Association;* Ernest Szekely, *National Association of Fan Manufacturers;* E. C. Webb, *Stoker Manufacturers Association;* C. M. Barber, *American Institute of Architects;* and H. A. Erf, *Acoustical Society of America.*

Management of the Exposition and all details of exhibit arrangement will be in personal charge of C. F. Roth of the International Exposition Co., New York.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the ASHVE Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, ASHVE Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.
F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

Fluid Flow. *Industrial and Engineering Chemistry,* Vol. 31, No. 4, April 1939, pp. 407-486. Twelve papers:—Introduction to Fluid Flow, by Warren L. McCabe; Selection of Pumps for Chemical Service, by Ward E. Pratt; Turbulence and Diffusion, by Hugh L. Dryden; Pressure Drop Accompanying Two-component Flow through Pipes, by L. M. K. Boelter and R. H. Kepner; Liquid Holdup and Flooding in Packed Towers, by J. C. Elgin and F. B. Weiss; Reduction of Shock Pressure in Solvent Delivery Lines, by Howard S. Gardner and John H. Folwell; Correlation of Rotameter Flow Rates, by John C. Whitewell and David S. Plumb; Eddy Diffusion, by W. L. Towle and T. K. Sherwood; Effect of A Screen Grid on the Turbulence of an Air Stream, by W. L. Towle, T. K. Sherwood and L. A. Seder; Performance of Ejectors, by Lincoln T. Work and Vincent W. Haedrich; Fluid Resistance in Pipes, by M. P. O'Brien, R. G. Folsom and Finn Jonassen; Concentric-cylinder Motor-driven Viscometer, by Wilhelm and Wroughton.

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Recent Developments in Pipe Protection, by C. H. S. Tupholme. *Colliery Engineering,* Vol. 16, No. 181, March 1939, pp. 101-02. Pipe corrosion in underground pipe systems depends on local factors of installation, as soil conditions and drainage, and upon uniformity of the pipe and the surface finish of the pipe. Explanation of cathodic protection. The use of coating materials for protection and some methods of applying them.

•

Typical Analysis of Water Hammer in a Pumping Plant of the Colorado River Aqueduct, by R. M. Peabody. *ASME Transactions,* Vol. 61, Feb. 1939, pp. 117-124. Method of analysis of surge conditions in a pumping system by arithmetical integrations, an extension of the work by Knapp. Analysis utilizes step-by-step method, includes complete characteristics of pumping system, and determines their effect upon surge conditions. Valve characteristics included.

•

Loss of Heat from the External Surface of a Hot Pipe in Air, by A. Bailey and N. C. Lyell. *Engineering,* Vol. 147, Jan. 20, 1939, pp. 60-62. Description of an investigation to determine the heat loss from the external surface of a horizontal pipe in a series of temperatures extending to 1000 F.

•

Rates of Water Vapor Adsorption from Air by Silica Gel, by J. Elston Ahlberg. U. S. Naval Research Laboratory, Washington, D. C. Report of experimental study of rates of adsorption of a condensable vapor contained in a non-condensable gas by a granular adsorbent. Silica gel and air containing variable amounts of water vapor were used.

•

Trends in Air Conditioning, by Howard E. Degler. *Plumbing and Heating Journal,* Vol. 106, No. 5, May 1939, pp. 30, 31 and 58. Discussion of smaller air conditioning systems, of attic fans for night cooling and of ice air conditioning systems.

•

The Design of a Central Warming Scheme for a Block of Flats in an Eastern Country, by Louis J. Overton. *The Heating and Ventilating Engineer,* Vol. XII, No. 143, May 1939, pp. 499-501. Discussion of heating necessary in tropical dwellings, and a system suitable for heating a block of flats which will satisfy such conditions.

Climatic Data for the Design and Operation of Air Conditioning Systems in Florida, by N. C. Ebaugh and S. P. Goethe. *Heating and Ventilating,* Vol. 36, No. 5, May 1939, pp. 37-42. Data for both winter and summer air conditioning: (1) dry-bulb temperature extremes and normals; (2) wet-bulb temperature extremes and normals; (3) degree-days of heating season and degree-hours of cooling season.

•

Equivalence of Coal to Oil for Heating Large Apartment Buildings in New York. *Heating and Ventilating,* Vol. 36, No. 5, May 1939, pp. 46-47. Report of tests made on heating large apartment buildings. Sixteen large income properties were heated, eight with coal and eight with oil, under similar conditions. One hundred seventy-five gallons of oil were found to be equivalent to one ton of anthracite coal.

•

Thermostatic Heat Control, by F. G. Czechowicz. *Coal Heat,* Vol. 35, No. 5, May 1939, pp. 5-7. How the active retail merchant can sell thermostatic heat control equipment to advantage.

•

The Advantages for an Atmosphere Control Room of a Quasi-continuous Record of Oxygen and Carbon Dioxide, by G. M. Bullowa and Grace Lubin. *The American Journal of the Medical Sciences,* Vol. CLXXXI, April 1931, No. 4, pp. 460-467. Description of apparatus for measuring oxygen by continuous flow of air. The thermoconductivity instrument records on a single timed chart at intervals of one minute both the percentage concentration of oxygen and the per mille concentration of carbon dioxide.

•

Ultraviolet Air Sanitation, by Frederick W. Robinson. *Industrial and Engineering Chemistry,* Vol. 31, No. 1, Jan. 1939, pp. 23-26. Discussion of sterilization of air by ultra-violet light. Bacterial efficiency for ultra-violet light of various wave lengths. Method of installing lamps in operating rooms and other wards of a hospital.

•

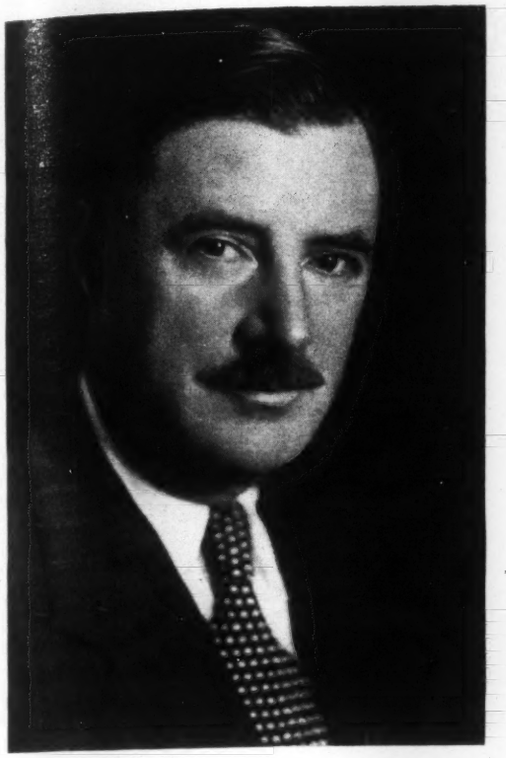
The Effectiveness of Certain Types of Commercial Air Filters Against Bacteria, by J. M. Dalla Valle. *Public Health Reports U. S. Public Health Service,* Vol. 54, No. 17, April 28, 1939, pp. 695-699. Description of a new method for the determination of the efficiency of air filters. Diagram of equipment and methods of operation.

•

Water Cooling Tower Fundamentals, by Edward Simons. *Chemical and Metallurgical Engineering,* Vol. 46. Series of three articles: The Theory of Atmospheric Cooling of Water, Feb. 1939, p. 85; Details of an Empirical Method for Designing Atmospheric Cooling Towers, Mar. 1939, pp. 146-149; Method for Computing the Slat-surface Requirements of a Mechanical draft Tower Employing a New Type of Packing (Bibliography), April 1939, pp. 208-211 and 217.

•

Structural Problems Attending Domestic Heating with Particular Reference to the Use of High Efficiency Stoves and Cookers, by L. W. Burridge. *Journal of the Royal Institute of British Architects,* Vol. 46, Series No. 11, April 3, 1939, pp. 559-561. General discussion of some structural problems. Explanation of effect of chimney difficulties caused by sulphuric acid gas in flue gas.



Pres. J. F. McIntire, Detroit, Mich.

Proceedings of the Semi-Annual Meeting 1939

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

The Grand Hotel, Mackinac Island, Mich.

July 4-6, 1939

MEMBERS of the Society registered from 20 states in the United States and Canadian provinces at the Semi-Annual Meeting 1939, held in the Grand Hotel, Mackinac Island, Mich., July 4-6. Exceptional interest in the technical sessions was evident and those who attended count this Great Lakes Meeting outstanding in the history of Society meetings. Members and their families were delighted with the friendly spirit and the splendid hospitality of the host Chapters and greatly enjoyed the delightful program provided by the Committee on Arrangements.

FIRST SESSION—

Wednesday, July 5, 9:00 a. m.

Pres. J. F. McIntire, Detroit, called the first session of the Semi-Annual Meeting 1939 to order. O. D. Marshall, Grand Rapids, Chairman of Activities on Mackinac Island, welcomed the Society members on behalf of the Michigan and Western Michigan Chapters and briefly outlined the program that had been prepared for their enjoyment.

President McIntire responded on behalf of the members and thanked Mr. Marshall and the Committee on Arrangements for the splendid program that had been prepared. Mr. McIntire then gave a brief outline of the activities that had been undertaken by the Society since the Annual Meeting and gave a resumé of his observations while visiting local Chapters in the south, on the west coast, and in the middle west. He gave some statistics about the Society's financial condition, the election of new members, distribution of THE GUIDE, and commented on some of the important work assigned to special committees.

The first technical paper was given by Professor Boelter of the University of California, Berkeley, and was entitled, The Effect of Vibration Upon Free Convection from Horizontal Cylinders by R. C. Martinelli and L. M. K. Boelter. (Complete paper published in August 1939 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Comments on the paper were made by H. M. Hart, Chicago, and Prof. F. J. Linsenmeyer, Detroit.

R. K. Thulman, Washington, D. C., co-author with Prof. L. E. Seeley, New Haven, Conn., of the paper, Heating Low-Cost Homes, made the presentation. (Complete paper published in August 1939 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

Some discussion was given by E. K. Campbell, Kansas City, Prof. C. M. Humphreys, Pittsburgh, C. Tasker, Toronto, H. M. Hart, Chicago, and C. F. Mally, Detroit. The prepared discussions of C. E. Mayette, New York, and T. H. Urdahl, Washington, D. C., were read by P. D. Close, Chicago, Ill., comments by W. E. Stark, Cleveland, were given by C. F. Cushing, Cleveland, and those of J. C. Fitts and H. C. Meyer, Jr., of New York, were presented by Professor Humphreys. Mr. Mayette believed that good progress had been made in coordinating engineering work for mass housing systems. Mr. Urdahl's opinion was that a more tangible interpretation of the value of the Society's work for general public good was clearly brought before the membership by the authors, and he expressed the thought that the paper was helpful in giving direction to necessary elements of research, standardization, and improvement in heating facilities. Mr. Meyer outlined the results of analysis of the heating methods used for sev-

eral large projects and his conclusion was that much had been accomplished in the matter of providing economical heating for low-cost housing, and much can still be done toward reduced cost of installation and operation. Mr. Fitts felt that the subject of heating for low-cost housing should be a matter of more general discussion by heating engineers and complimented the authors on their viewpoint that operating cost is of even more importance than the initial cost of the system. Mr. Stark believed that greater protection should be offered to owners of low-cost homes to insure adequate heating equipment and recommended that the Society take a leading role in rectifying present unsatisfactory conditions which had been pointed out by the authors.

The last paper at the morning session, entitled Evaluating Performance of Water Heaters Fired with Solid Fuels by H. J. Rose and R. C. Johnson, was given by Mr. Rose, Pittsburgh. (Complete paper published in July 1939 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

Comments on the paper were made by E. K. Campbell, Kansas City, H. M. Hart, Chicago, and W. J. Whelan, Detroit. A prepared discussion was presented by A. J. Johnson, Primos, Pa. Mr. Rose read the comments of E. T. Selig, Jr., Pittsburgh, who commented on the variable nature of residential demands for domestic hot water and urged the necessity of designs which place greater emphasis on performance requirements for user satisfaction and convenience rather than on rating and output of the units. Mr. Johnson commented on the inadequacy of present-day rating methods and recommended a complete revision of the system of rating domestic heaters. W. A. Danielson, Ft. Knox, Ky., urged the establishment of new rules for testing water heaters fired with solid fuels and recommended that manufacturers include a name-plate on all equipment giving essential information on the output of the unit.

Messages of greeting from W. J. Olvany, New York, president of the *Heating, Piping and Air Conditioning Contractors National Association*, and A. W. Williams, Columbus, managing director of the *National Warm Air Heating and Air Conditioning Association*, were read.

SECOND SESSION—

Wednesday, July 5, 1:30 p. m.

The second session was called to order on Wednesday afternoon, July 5, and the paper, Ozone in Ventilation—Its Possibilities and Limitations by W. N. Witheridge and C. P. Yaglou, was presented by Mr. Witheridge of Detroit. (Complete paper to be published in ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

Dr. M. B. Ferderber, Pittsburgh, presented written comments on his observations of the use of ozone as a de-odorizer and quoted the results of research in the production and use of ozone by H. F. Smyth, Jr., industrial fellow, Mellon Institute.

E. K. Campbell, Kansas City, Mo., chairman of the Membership Committee, was introduced and presented his report on the status of Society membership.

Report on Membership

Out in Kansas there is a group of country editors who have made quite a reputation for their wit and humor. One of them was asked why it was that a married man seemed to get further than a bachelor and his reply was, "No man runs his best unless there is someone after him."

I think it was on that theory that the Council appointed the Membership Committee with a Chairman whose business it is to keep after the Chapters and their Membership Chairmen, and it is not an easy job.

Giving the statistics on membership, we now have a total of 3127 members of all grades. During the first six months of 1939 we received 215 applications as compared to 237 applications in the first six months of 1938. During the same period 214 members have been elected, whereas 554 were elected in the same period of 1938. This large number in the first six months of 1938 was due to the fact that many applications were received during December, 1937, because on the first of January the $10 initiation fee again became effective.

During the first six months of 1939 there was a loss of 190 members through resignation, cancellation or death; during the corresponding period in 1938 there was a loss of 193. During the second six months period of 1938 there were not so many as only 20 were dropped. Hence, if we can keep the membership activity going during the balance of this year, we may be able to make a fair showing. Against the number dropped in the first six months of this year (190), 214 members have been actually elected, making a gain of 24. The Committee hopes for considerable more gain in the second six months.

There seem to be two theories or two schools of thought regarding the Membership Committee's activities. I have regarded the Chairmanship of the Membership Committee as that of a sales manager with a crew of salesmen, selected by someone else and over whom I have little or no authority, but I have regarded the work of obtaining new members strictly as a sales proposition. We have something to sell.

On the other hand, some Chapters and some members have considered that active attempts to "sell" membership in the Society have been beneath their dignity, and the dignity of the Society. This has had a very natural result in that the Chapter which particularly upholds that idea has the smallest membership in proportion to population of all of the Chapters in the Society.

In my judgment it makes no difference how good a thing a person has to sell or give away, if he does not let anyone know about it he is neither going to sell nor give away very much. The world makes a beaten path to the door of the best mouse trap, only when the world knows about it. If we wait in our membership work for someone else to spread the news we are not going to sell or give away or share the privilege of membership in this Society to any great extent.

I regard it as a business of the Membership Committee and of the Chapter Membership Committees to spread the news, to sell membership. This Society is doing a great work, and I know of no other similar Society doing as good work, and as important work as our Research. Our Research program has grown until, including both sides of our cooperative programs it represents an expenditure in the neighborhood of $100,000 annually. Our research work has developed many facts that were in the cloudy area when we began this work. The results of its work have been far reaching.

So we have something to sell and we have something to sell that is worth buying on the part of prospective customers. We are giving value received whenever we sell a membership, and any man engaged in any of the allied industries, receiving as he does material benefits from this Society, should be willing to become a member if for no other reason than that he is willing to help.

Having this thought in mind, you will find, as I have found, that many times it is necessary only to suggest to someone connected with the industry or the profession that it would be to his interest to become a member, and that is all that is

required. If each member of the Society (and I wish you would take this message back to your Chapters) will simply carry it in his mind to give men of the right sort an opportunity to become members, you will be surprised at the number of new members who will join.

E. K. CAMPBELL, *Chairman*
MEMBERSHIP COMMITTEE

The subject, Effect of Size and Type of Air Inlet and Outlet on the Heat Output of Convectors by A. P. Kratz, M. K. Fahnestock and E. L. Broderick was given by Professor Fahnestock, Urbana, Ill. (Complete paper to be published in ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

Oral comments were offered by E. Holt Gurney, Toronto, H. M. Hart, Chicago, Prof. F. B. Rowley, Minneapolis, and H. J. Young, Muskegon, Mich. In a written discussion, W. A. Rowe, La Crosse, Wis., commented on the difficulty of dealing with small air quantities and the problem of getting appreciable capacity increases and he expressed the hope that the limited number of tests would be expanded. R. H. Norris, Schenectady, believed that the results obtained so far were of great practical value and suggested that further information would be desirable on the relative flow resistances of the different heating units used in the tests. J. W. McElgin, Philadelphia, suggested that the authors investigate the influence of the length of outlet on the capacity top outlet convector, as experience indicated that the physical length of the outlet in relation to both active convector length and cabinet length is of major importance. In a joint written discussion by Ernest Szekely and C. H. Randolph, Milwaukee, the suggestion was made that three important phases of the problem receive study, the effect of stack height, heating mediums other than steam, and effect of baffling both inside and outside of the convector.

Prof. F. B. Rowley, Minneapolis, presented his paper on A Theory Covering the Transfer of Vapor Through Materials. (Complete paper published in July 1939 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

Oral comments were made by A. E. Stacey, Jr., New York, Prof. W. T. Miller, Lafayette, Ind., Prof. L. M. K. Boelter, Berkeley, Calif., and C. F. Mally, Detroit. In a written discussion, E. C. Lloyd, Lancaster, Pa., mentioned three items for further exploration and commended the author for his contribution toward the solution of problems presented by condensation in insulation and proposed that actual application of these theories be made by examining the performance of various types of structures under highly humid operating conditions. R. H. Heilman, Pittsburgh, felt that the paper gave a more complete understanding of vapor transfer and its effect upon the condensation of moisture in building walls. Wharton Clay, New York, favored the wider use of non-hygroscopic insulating materials and suggested humidity limits for low temperature conditions in order to prevent condensation troubles. W. A. Danielson, Ft. Knox, Ky., said it is now possible for a designer to plan for proper insulation where vapor pressures differ on two sides of a wall and he described the construction of a cold storage plant to show the effectiveness of a vapor barrier. W. V. Hukill, Washington, D. C., desired to know if any tests had been made to demonstrate the apparent reversal in the direction of moisture content and vapor pressure gradients and hoped that a future paper giving a fuller discussion of the theory of such flow and of the laws by which the direction of moisture travel is evident would be presented, which would be helpful to a better understanding of the problem. P. F. McDermott, Manville, N. J., presented some ideas regarding the phenomenon of hygroscopicity and commented on the necessity for determining the resistance of building structures.

The final paper of the session was presented by R. C. Loughead, Detroit, the author, on the subject of Fire Protection for Air Conditioning Systems. (Complete paper to be published in ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

Those who discussed the paper were Prof. C. M. Humphreys, Pittsburgh, E. K. Campbell, Kansas City, J. B. Hewett, New York, H. M. Hart, Chicago and Professor Rowley, Minneapolis. In a written discussion, H. C. Murphy, Louisville, Ky., suggested that heating and ventilating engineers cooperate closely with fire protection engineers and offered some comments on the fireproofing of air filters and the need for regular servicing and inspection of equipment.

THIRD SESSION—

Thursday, July 6, 9:00 a. m.

The final session of the Semi-Annual Meeting 1939 was opened on Thursday morning, July 6, by Pres. J. F. McIntire and he introduced F. C. Houghten, director of the Society's Research Laboratory, Pittsburgh, who presented the paper, Summer Cooling Requirements in Washington, D. C. and Other Metropolitan Districts by F. C. Houghten, Carl Gutberlet and A. A. Rosenberg. (Complete paper to be published in ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

Comments from the floor were made by Dr. M. B. Ferderber, Pittsburgh, Prof. F. J. Linsenmeyer, Detroit, and C. Tasker, Toronto. Written discussions prepared by M. G. Kershaw, Wilmington, Del., T. H. Urdahl, Washington, D. C., and Dr. C. A. Mills, Cincinnati were read by John James, Technical Secretary, New York. Mr. Kershaw cited personal experiences and observations in the operation of air conditioning systems and said that shock could be avoided by raising inside temperatures during the afternoon in order to lower the differential between inside and outside conditions. He expressed the opinion that a sliding scale effective comfort line should be developed. Mr. Urdahl commented on the significance of the data collected in order to obtain exact knowledge of the temperatures at which most people attain the greatest sensation of comfort. Dr. Mills urged closer control of the subjects used in the studies from a physiological standpoint and pointed out the factors that must be thoroughly studied before summer cooling and winter heating tests can be successfully carried out because of man's variable heat producing mechanism.

President McIntire introduced W. L. Fleisher, New York, chairman of the Committee on Research, who summarized the report of the committee as outlined.

Report on Research Activities

W. L. Fleisher, chairman of the Committee on Research, spoke of several outstanding accomplishments during the first six months of 1939, one being the consummation of an agreement with the U. S. Navy Department, Washington, D. C., which involves a comprehensive cooperative program extending over a period of two years for which the government will contribute $10,000 per annum. The program is under the direction of a special committee composed of W. L. Fleisher, J. H. Walker, C.-E. A. Winslow, A. E. Stacey, Jr., and Director F. C. Houghten.

Projects at Pittsburgh

(1) From a contribution of $3,500 in earmarked funds, test cubicles to study the solar radiation transmittance through glass block wall construction have been erected at the Laboratory. Because this study is of vital interest to engineers concerned with the application of summer cooling and air conditioning in buildings containing glass block construction, the results will be of wide interest. The work is proceeding under the supervision of the Technical Advisory Committee on Air Conditioning Requirements of Glass, of which M. L. Carr is chairman.

(2) A fund of $6,000 has been collected for studies outlined by the Technical Advisory Committee on Radiation and Comfort, of which J. C. Fitts is chairman. Two test rooms have been erected on the roof of one of the Laboratory buildings in Pittsburgh and arranged with similar exposures in order that the effect of radiant heat may be appraised on a comparative basis with other commonly accepted methods of heating.

(3) Studies of the flow of air in ducts continue under the direction of Chairman J. H. Van Alsburg and his Technical Advisory Committee on Air Distribution and Air Friction. The present work involves round elbows and their square equivalent after which investigations will be conducted on straight rectangular ducts.

(4) Chairman C. S. Leopold and his Technical Advisory Committee on Cooling Load in Summer Air Conditioning have been instrumental in securing funds for the erection of two test buildings to determine the heat flow through various built-up wall sections.

(5) The physiological reactions of subjects to various heat outputs in summer and winter are being studied by the Technical Advisory Committee on Air Conditioning in Industry, of which A. E. Stacey, Jr., is chairman.

(6) Data analyzed by the Laboratory staff for summer cooling requirements in different metropolitan districts are presented in a paper at this meeting. Under the direction of C. Tasker and his Technical Advisory Committee on Sensations of Comfort, approval has been given for studies outlined by a Joint Committee of the ASHVE and the *Illuminating Engineering Society* which will be conducted at the Laboratory to determine human reactions to color sensations in various environmental conditions.

Cooperating Institutions

Research under cooperative agreements is being conducted at the following colleges:

University of California. Project outlined by the Technical Advisory Committee on Cooling Towers, Evaporative Condensers and Spray Ponds, of which B. M. Woods is chairman.

Case School of Applied Science. Two projects dealing with air distribution are under investigation. Another study comes under the direction of the Technical Advisory Committee on Heat Transfer of Finned Tubes with Forced Air Circulation, of which G. L. Tuve is chairman.

University of Illinois (Engineering—Urbana). This investigation is concerned with the effect of inlet and exhaust opening location on the motion and distribution of air in a room.

University of Illinois (Medical—Chicago). Data are being collected on the determination of cardiac output, blood volume, skin temperatures and oxygen saturation of venous blood at comfortable, hot, dry and wet conditions for various subjects.

Lehigh University. Commencing July 1 an investigation started on round and square elbows to determine resistance offered to air flow.

Massachusetts Institute of Technology. Studies are planned at this institution on the attenuation of sound through lined and unlined ducts. This is one phase of the program of the Technical Advisory Committee on Sound Control, J. S. Parkinson chairman.

Michigan College of Mining and Technology. Through the Technical Advisory Committee on Corrosion a program has been developed at this institution under the direction of A. R. Mumford who is chairman of this committee dealing with the determination of the degree of contamination of condensate formed at varying rates from steam of known composition.

University of Minnesota. Investigation of air filter performance and methods of rating air filters is being continued under a renewal agreement effective July 1. A suitable field test is also being studied.

University of Pennsylvania. This research contemplates an experimental investigation of the departures from Dalton's law in the case of moist air and has been planned by the Technical Advisory Committee on Psychrometry of which F. R. Bichowsky is chairman.

University of Pittsburgh. Studies at the Elizabeth Steel Magee Hospital deal solely with the bacteriological content of the air in an air conditioned operating room and have been supervised by the Technical Advisory Committee on Treatment of Disease, of which Dr. M. B. Ferderber is chairman.

Agricultural and Mechanical College of Texas. This research under the direction of the Technical Advisory Committee on Heat Requirements of Buildings, of which P. D. Close is chairman, is intended to ascertain flow of heat through walls for both heating and cooling cycles.

University of Wisconsin. This is another study which comes under the supervision of the Technical Advisory Committee on Air Distribution and Air Friction.

Advisory Committees' Activities

Committee on Air Pollution and Purification. Realizing that the membership of the Society has a definite interest in the rather broad field of air pollution and purification the Research Executive Committee approved the appointment of C.-E. A. Winslow to act as general chairman for the ensuing year. An invitation was extended to H. B. Meller to act as chairman of the Sub-Committee on Air Pollution and Dr. Leonard Greenburg has accepted the invitation to become chairman of the Sub-Group on Removal of Atmospheric Impurities. The personnel of this general committee has just been completed and it is expected that in the near future a program of activity will be outlined and initiated.

Committee on Instruments. This committee was formed with D. W. Nelson chairman, and its program will be announced shortly.

Committee on Weather Design Conditions. Chairman T. H. Urdahl has been setting up a program in cooperation with the various utility organizations to determine uniformly outdoor dry- and wet-bulb temperatures in the principal cities.

Committee on Solid Fuels. Chairman W. A. Danielson has designated several sub-groups to report on anthracite, bituminous and sub-bituminous coals as well as lignite and coke. Another sub-committee on chimneys has been very active since the first of the year in outlining a program of research.

Committee on Insulation. R. T. Miller, chairman, reports co-operative work under way with the Guide Publication Committee and development of a chart showing the recommended relative humidities to be maintained inside the buildings with varying outdoor conditions.

The paper, Study of Summer Cooling in the Research

Residence for the Summer of 1938 by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick, was presented by Professor Kratz, Urbana, Ill. (Complete paper published in August 1939 ASHVE JOURNAL SECTION, *Heating Piping and Air Conditioning*).

Written discussions by L. T. Avery, Cleveland, A. B. Newton, Minneapolis, and Mr. Urdahl were read by Mr. James, and oral comments were made by C. F. Cushing, Cleveland, and Prof. F. B. Rowley, Minneapolis. Mr. Avery believed the authors' investigations would be helpful in extending the use of simple and inexpensive cooling systems for medium sized residences. He also commented on the contrast between intermittent operation and continuous operation of cooling plants in commercial and residential work, but warned that summer air conditioning will receive public acceptance only when the design and the equipment are adequate to assure satisfactory comfort. Mr. Newton expressed particular interest in the graphical comparison between the results obtained with the different size condensing units and he suggested a means of reducing the *on* and *off* cycles and proposed an extended series of comparative tests for future consideration. Mr. Urdahl expressed particular interest in the authors' observations on heat flow and lag through insulated and non-insulated walls, and he expressed the hope that continued research would provide a wider application of air conditioning for summer comfort with initial and operating costs much below present levels. He gave several examples of air conditioning system performance that checked closely with the results obtained by the authors.

Vice-Pres. F. E. Giesecke, College Station, Tex., took the chair and introduced Prof. L. M. K. Boelter, who gave his paper, Cooling Tower Performance Studies. (Complete paper to be published in ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Vice-President Giesecke expressed the appreciation of the members for the papers and inquired if there was any unfinished business to come before the Society.

As nothing was offered, he called for the report of the Resolutions Committee, which was read by Secy. A. V. Hutchinson, New York, and was unanimously adopted.

Report of Resolutions Committee

The members of the Society attending the Semi-Annual Meeting 1939 at Mackinac Island, July 4, 5 and 6, having enjoyed to the utmost the thoughtful and complete arrangements which have contributed so much to the delightful meeting now coming to a close, wish to express their thanks and appreciation as follows:

To W. G. Boales, general chairman, and his able and hard working Committee on Arrangements of the Michigan and Western Michigan Chapters for their untiring efforts in our behalf;

To the membership of the two Chapters as a whole for their splendid cooperation in their joint sponsorship of the meeting;

To Emery Hatch and staff of the Georgian Bay Line for the courteous assistance rendered those members who came by boat;

To W. S. Woodfill and every employee of The Grand Hotel whose unfailing thoughtfulness and courtesy have contributed so largely to our pleasure;

To Col. Roger Andrews, whose all too brief talk on this historic spot at the semi-annual banquet was outstanding. It was not only a highlight of our meeting, but it also aroused in all of us a desire for further knowledge about this Hill of History;

To the trade press and newspapers who devoted so much space to recording the events of this meeting and to the *Mackinac Island News* for the gracious write-up in the July 1 issue, and finally,

To the authors of the papers presented at this meeting for their patience in collecting the necessary data and the time required to present it in proper form;

Be It Therefore Resolved, That the above is the sentiment of this Semi-Annual Meeting and that a copy of these resolutions be sent to each of the above named individuals and organizations.

Respectfully submitted,

R. H. CARPENTER, *Chairman*
H. H. ERICKSON
W. A. RUSSELL

President McIntire assumed the chair and made the concluding announcements, after which the Semi-Annual Meeting 1939 adjourned.

Council Meetings

At Mackinac Island on July 4 and 6, the Council held two meetings. It received reports from the Executive, Finance, Meetings, Membership and Standards Committees, and a number of special committees including the F. Paul Anderson Award, Educational Publicity, Regional Meetings, Engineering Scholarship, Constitution and By-Laws, and Engineering in its Relation to Public Health.

A petition from Portland, Oregon, members requesting a charter for a Chapter in Portland was approved.

Nomination of five members to serve on the Committee on Research for a three-year term beginning 1940 was confirmed and will be published in the October JOURNAL.

Research Committee

The Committee on Research met at Mackinac Island on July 4 with 12 members and several officers of the Society and guests in attendance. Cooperative agreements were extended with the University of Wisconsin and Case School of Applied Science for investigations on air distribution and with the University of Minnesota for a program of air filter research to be approved by the Technical Advisory Committee on Air Pollution and Purification.

Several reports were received, including that of the Finance Committee, on the possibility of cooperation with the U. S. Bureau of Standards, and the Air Conditioning Committee of Edison Electric Institute. Director F. C. Houghten reviewed a comprehensive outline of Society research, which he had prepared, and summarized projects at the Laboratory in Pittsburgh.

Discussion was given to the Proposed Regulations Governing ASHVE Research Technical Advisory Committees, which are to be sent to members of the Committee on Research for approval by letter ballot.

W. H. Carrier Entertained at Washington, D. C.

The officers and board of governors of Washington, D. C. Chapter were guests of T. H. Urdahl at the Cosmos Club on Saturday afternoon, June 24, to meet W. H. Carrier, past president of the ASHVE. Mr. Carrier made a brief visit to Washington before leaving on a trip to South Africa. Maurice Firestone, manager of the Carrier Division of the United Clay Products Co., and other friends were among those present.

Fun for Everyone at Mackinac

An extensive entertainment and sports program was provided for members and guests attending the Society's Semi-Annual Meeting at Mackinac Island, July 4-6, and all activities were under the general supervision of O. D. Marshall, Grand Rapids, Chairman of Activities on the Island. A delegation from the Reception, Transportation and Ladies Committees was at the boat pier with the Hotel Orchestra when the S. S. South American arrived with a group of 110 who had taken the lake cruise en route to the meeting. Passengers and baggage were brought to the Grand Hotel in the horsedrawn conveyances which are typical of Mackinac Island.

Registration started July 4 at 8:30 a. m. and sports activities were the immediate objective of many, while other members attended Council and Committee meetings which were held throughout the day.

At 9:30 a. m. an 18-hole medal play golf tournament was held on the hotel course with 50 players participating. Competition was for the Research Cup and the Eichberg Memorial Cup. G. D. Winans, Detroit, Chairman of the Golf Committee, was in charge.

On Wednesday afternoon, July 5, a ladies golf tournament was held and those who participated were Mrs. H. C. Murphy, Mrs. E. Holt Gurney, Mrs. J. H. Milliken, Mrs. S. F. Myers, and Mrs. R. W. Schneider.

The golf tournament winners were as follows:

Research Cup—J. H. Milliken, Chicago.
Eichberg Cup—Illinois Chapter team (Tom Brown, J. H. Milliken, R. E. Moore).
Low Gross—M. F. Blankin, Philadelphia.
Runner-up—Tom Brown, Chicago.
High Gross—F. R. Bishop, Detroit.

Winners of the kicker's handicap were:

E. W. Rietz, Chicago,
W. M. Myler, Jr., Columbus, O.
T. A. Novotney, Johnstown, Pa.,
M. W. Bishop, Chicago,
N. D. Adams, Rochester, Minn.,
W. G. Schlichting, Kalamazoo,
A. E. Kriebel, Philadelphia,
C. E. Price, Chicago,
George Kingsland, Minneapolis,
H. H. Erickson, Philadelphia.

Special awards:

F. L. Meyer, Peoria, most birdies;
F. J. Linsenmeyer and G. H. Tuttle, Detroit, most 7's;
R. F. Hunger, Philadelphia, 100;
C. F. Cushing, Cleveland, low score 4th hole;
R. K. Milward, Detroit, low score 7th hole;
R. T. Miller, Chicago, most pars.

Guest prizes:

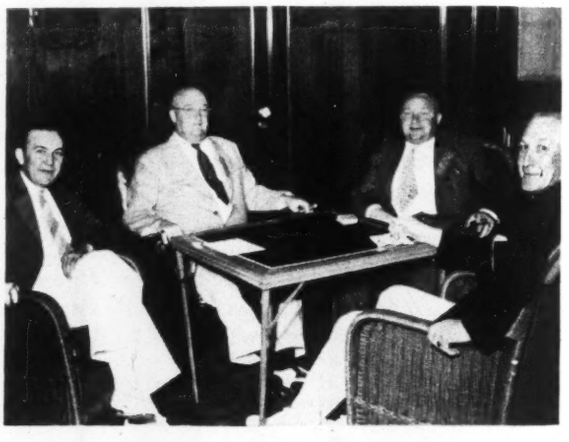
George Kelting, Minneapolis, low net;
R. J. Resch, Minneapolis, high score.

A sports program and children's games were handled by Messrs. B. F. McLouth and S. S. Sanford and during the afternoon the children enjoyed a hiking party and water sports at the pool. At 5:45 the children gathered for a bus ride around the island and greatly enjoyed the novelty of being driven in a horse-drawn carriage.

During the afternoon of July 4 a group of 48 ladies enjoyed a bridge game and afternoon tea. At 6:00 p. m. the Blue Room was the scene of an enjoyable social hour and over 100 members and guests participated. A special feature was the presentation of good luck horseshoes to the Officers and Council members by W. C. Randall, Detroit, President of the Michigan Chapter. This unique gift was a horseshoe that had come from Mackinac Island and had been polished and plated. A gift from the members of the Michigan and Western Michigan Chapters was presented to W. G. Boales, General Chairman of Arrange-

Allen Johnson and the Ferry sisters have a birdseye view

Time out for J. S. Kilner, W. G. Boales, and Fred Johnson



A quartette from Detroit—Frank Connell, Bob Milward, E. Anderson and W. J. Whelan

Fair foursome—Miss Helen Rose, Mrs. A. E. Stacey, Jr., Mrs. W. G. Boales and Mrs. J. H. Walker

Photos by Paul J. White

Merrill Blankin thinks it's fun, but it's serious business for Tom Urdahl and John Landers

ments by O. D. Marshall, Chairman of Activities on Mackinac Island.

After dinner on July 4th, many attended the civic celebration and enjoyed the folk dancing and fireworks. At 10:00 p. m. the crowd gathered in the Casino at the Grand Hotel and enjoyed a floor show and dancing. During the festivities Prof. F. J. Linsenmeyer of Detroit, Past President of the Michigan Chapter, presented a fine traveling bag to J. F. McIntire, President of the Society, on behalf of the Western Michigan and Michigan Chapters.

On Wednesday, July 5th, many of the ladies started on a bicycle trip around the Island and others enjoyed a carriage ride. At 11:00 a. m. S. H. Downs led a group of fishermen to the boat landing for a trolling trip. When they returned, it was conceded that S. F. Myers caught the largest fish and M. F. Blankin caught the most fish. Others who participated were: L. C. Soule, Syracuse, T. D. Stafford, Grand Rapids, H. J. Metzger, Kalamazoo, R. L. Blanding, Providence, Dr. M. B. Ferderber, Pittsburgh, E. J. Rodee, New Haven, N. D. Adams, Rochester, Minn., J. S. Kilner, Detroit, F. W. Johnson, Detroit.

At 2:15 p. m. 80 ladies and children embarked on the steamer, Mackinac Islander for a trip around the Island. At 6:00 p. m. the children enjoyed a beach party and picnic supper.

The Semi-Annual banquet and dance was held in the Casino of the Grand Hotel and at the conclusion of dinner, Prof. E. O. Eastwood, toastmaster, introduced Prof. F. B. Rowley who presented the Past President's Memory Book to E. Holt Gurney, Toronto. Mr. Gurney expressed his appreciation for the fine volume that had been compiled and which would remind him of many pleasant incidents during his service as President of the Society.

C. R. McConner, Kalamazoo, presented Mrs. J. F. McIntire with a beautiful bouquet of roses on behalf of the Michigan and Western Michigan Chapters.

J. H. Walker, Vice Chairman of the Committee on Research, presented the Research Cup to J. H. Milliken of Chicago and Tom Brown, President of the Illinois Chapter, received the Eichberg Cup from M. F. Blankin, Philadelphia.

A brief and interesting address on the Hill of History was given by Col. Roger Andrews, who has made a study of the historical data of Mackinac Island. Following this unusually interesting address, Professor Eastwood presented Pres. J. F. McIntire and also introduced A. F. Nass, a director of the *Heating, Piping and Air Conditioning Contractors National Association.*

Professor Eastwood announced the winners of the golf tournaments and introduced the members of the General Arrangements Committee and the members of the Michigan and Western Michigan Chapters. Dancing to the music of Charlie Zwick and his orchestra was enjoyed from 10:15 p. m.

On July 6th the ladies had a hiking party to Fort Mackinac and in the afternoon they played bridge. There were informal golf matches on the hotel and country club courses and at 3:00 p. m. those who came by boat from eastern cities, were escorted to the pier where they sailed on the S. S. South American. Those who made the trip by boat reported a most interesting trip with perfect weather and a most enjoyable time with their shipmates.

Members from 20 states made up the total registration of 257 and everyone agreed that the meeting was an outstanding success.

Dinner-time on the *South American* and the Carpenter family, the Moodys, Blandings, Blankins and Evans are enjoying it

Committee on Arrangements

W. G. Boales, *General Chairman*, Detroit

S. H. Downs, Kalamazoo — F. J. Linsenmeyer, Detroit
C. R. McConner, Kalamazoo — W. C. Randall, Detroit
B. F. McLouth, Lansing — J. H. Walker, Detroit

Finance:
J. S. Kilner, Detroit — C. R. McConner, Kalamazoo
F. J. Linsenmeyer, Detroit

Attendance: F. C. Warren, Grand Rapids, *Chairman*
C. H. Morton, Grand Rapids — L. L. McConachie, Detroit
E. M. Harrigan, Detroit

Greeters Committees

Detroit: G. H. Tuttle, *Chairman*
G. W. Akers — F. J. Feely
E. H. Clark — J. S. O'Gorman
W. C. Randall — M. B. Shea

Cleveland: D. L. Taze, *Chairman*
L. T. Avery — W. R. Rhoton

Buffalo: J. J. Landers, *Chairman*
M. C. Beman — C. W. Farrar
Joseph Davis — L. P. Saunders

Chicago: Tom Brown, *Chairman*
J. R. Vernon

S.S. North American:
R. E. Moore, Chicago — A. P. Kratz, Urbana
J. J. Aeberly, Chicago

S.S. South American:
F. J. Feely, Detroit — W. J. Whelan, Detroit
R. K. Milward, Detroit — G. H. Tuttle, Detroit
R. F. Connell, Detroit — F. W. Johnson, Detroit

Transportation in Detroit: M. B. Shea, *Chairman*
J. F. Lance — L. A. Burch

Bon Voyage Committee, Detroit: E. H. Clark, *Chairman*
M. B. Shea — L. A. Burch
C. L. Toonder

Canadian Trips:
W. J. Whelan, Detroit — F. J. Linsenmeyer, Detroit
R. K. Milward, Detroit — E. J. Anderson, Detroit
R. L. Deppmann, Detroit

COMMITTEES FOR MACKINAC ISLAND ACTIVITIES

O. D. Marshall, Grand Rapids, *Chairman*

Reception: R. F. Connell, Detroit, *Chairman.*
J. S. O'Gorman, Detroit — G. H. Tuttle, Detroit
M. W. Bishop, Chicago — S. W. Todd, Grand Rapids
F. C. Warren, Grand Rapids — S. H. Downs, Kalamazoo

Transportation: H. F. Reid, Muskegon Heights, *Chairman*
S. S. Sanford, Detroit, *Vice-Chairman*
J. S. Cunningham, Dowagiac — J. G. Dykman, Grand Rapids
W. H. Old, Detroit — P. J. Vanderlip, Lansing
H. D. Bratt, Grand Rapids — H. J. Young, Muskegon
W. F. Graff, Grand Rapids — S. J. Dempsey, Battle Creek

Publicity: T. D. Stafford, Grand Rapids, *Chairman*
S. L. Worthing, Grand Rapids — H. F. Hutzel, Detroit

Golf: G. D. Winans, Detroit, *Chairman*
W. G. Schlichting, Kalamazoo — R. L. Deppmann, Detroit
A. E. Knapp, Detroit — E. J. Anderson, Detroit

Children's Activities: S. S. Sanford, Detroit, *Chairman*
Mrs. F. J. Feely — Mrs. R. F. Connell
Mrs. J. H. Walker — Mrs. G. D. Winans
Mrs. A. E. Knapp — Mrs. S. H. Downs
Mrs. O. D. Marshall — Mrs. G. H. Tuttle

Sports: B. F. McLouth, *Chairman,* and Mrs. McLouth, Lansing
Mr. and Mrs. R. E. Distel, Lansing — Mr. and Mrs. V. H. Hill, Lansing

Fishing Trip: S. H. Downs, *Chairman,* Kalamazoo
H. J. Metzger, Kalamazoo — T. D. Stafford, Grand Rapids

Sessions: W. W. Bradfield, Grand Rapids, *Chairman*
W. C. Randall, Detroit, *Vice-Chairman*
R. S. M. Wilde, Highland Park
A. E. Knibb, Detroit — G. D. Woodhouse, Dowagiac

Ladies: H. F. Hutzel, Detroit, *Chairman*
B. F. McLouth, Lansing
Mrs. L. G. Miller, East Lansing — Mrs. O. D. Marshall, Grand Rapids
Mrs. C. R. McConner, Kalamazoo — Mrs. R. F. Connell, Detroit
Mrs. L. L. McConachie, Detroit — Mrs. Tom Brown, Chicago

Special Boat Trips: C. H. Pesterfield, East Lansing, *Chairman*
E. J. Anderson, Detroit — S. S. Sanford, Detroit
B. F. McLouth, Lansing — R. K. Milward, Detroit

Banquet: L. G. Miller, East Lansing, *Chairman*
F. R. Bishop, Detroit — A. Marin, Ann Arbor
W. C. Randall, Detroit

MICHIGAN CHAPTER

President: W. C. Randall — *Secretary:* W. H. Old
Vice-President: G. H. Tuttle — *Treasurer:* J. S. Kilner
Board of Governors: J. S. O'Gorman, S. S. Sanford, M. B. Shea

WESTERN MICHIGAN CHAPTER

President: B. F. McLouth — *Secretary:* C. H. Pesterfield
Vice-President: T. D. Stafford — *Treasurer:* C. H. Morton
Board of Governors: C. R. McConner, W. G. Schlichting, F. C. Warren

Officers Elected by Montreal Chapter

May 22, 1939. Montreal Chapter met at the Windsor Hotel with an attendance of 39 members and guests. The meeting was called to order by Pres. F. J. Friedman at 8:00 p. m. and minutes of the previous meeting were read by Secy. C. W. Johnson and accepted by the meeting.

Annual reports were submitted by the president, secretary and treasurer. Reports for the year were also submitted by the Membership Committee and the Attendance Committee.

The next order of business was the report of the tellers. Officers elected for the ensuing season 1939-40 were as follows:

President—L. H. Laffoley.
Vice-President—W. U. Hughes.
Secretary—C. W. Johnson.
Treasurer—G. E. Perras.
Board of Governors—A. B. Darling, G. L. Ballantyne, F. G. Higgs, F. J. Friedman, G. L. Wiggs.

The speaker of the evening, M. B. McLachlan, was introduced by Mr. Friedman, and presented his Personal Experiences of a British Secret Service Agent During the Great War. A vote of thanks was extended to the speaker on behalf of the Chapter by Mr. Ballantyne.

The chair was then turned over to the incoming president, Mr. Laffoley, by Mr. Friedman, past president.

New business brought before the Chapter included a discussion of a June golf meet. A motion was made by Mr. Ballantyne and was seconded and carried that the Golf Committee be given authority to select the date and place.

The meeting was adjourned at 11:30 p. m. according to the report of Secretary Johnson.

Western New York Annual Meeting

May 10, 1939. The annual meeting of Western New York Chapter was held at the University Club, Buffalo, with 23 members and guests attending.

Pres. J. J. Landers appointed H. G. Zolitsch, Herman Seelbach and H. A. Ruff as tellers. A count of the 30 votes cast for the officers and board of governors for 1939-40 gave the following results:

President—L. P. Saunders.
1st Vice-President—C. A. Gifford.
2nd Vice-President—W. R. Heath.
Secretary—H. C. Schafer.
Treasurer—B. C. Candee.

The following were declared elected to the board of governors: D. J. Mahoney, M. C. Beman, Joseph Davis, Roswell Farnham and J. J. Landers, retiring president, automatically becomes the fifth member of the Board of Governors, under the amendment to the Constitution adopted in 1937.

The Chapter voted $25.00 towards the expenses of the delegate, Mr. Landers, to the Society's Semi-Annual Meeting at Mackinac Island, Mich.

The treasurer's report for the fiscal year was presented by Mr. Schafer and showed a balance as of May 8 of $163.91.

The auditing committee reported that the records of Treasurer Schafer were in good order and the amount as reported was correct.

C. W. Farrar urged that every member make an effort to attend the Semi-Annual Meeting at Mackinac Island, July 4-6.

Mr. Landers then turned the meeting over to President-elect Saunders, and later the chair was occupied by Mr. Gifford. B. C. Candee presented a past president's certificate to the retiring president and, in accepting this, Mr. Landers expressed his appreciation for the support which the officers and members gave him during his administration.

Vice-President Gifford then introduced the speaker of the evening, T. L. Wilkinson, district PWA supervisor, who had chosen as his subject the Workings of the PWA. Mr. Wilkinson gave a very interesting and instructive talk after which he was elected an honorary member of the Chapter.

C. A. Bulkeley Dies

It is with sorrow that the unexpected death of Claude A. Bulkeley must be recorded on Sunday, July 9, 1939, at Idylease Sanatorium, Newfoundland, N. J., where he had been ill for a number of weeks. Funeral services were conducted on July 12 at Iliff Funeral Home, Newton, N. J., the town in which his summer home was located. Masonic services were conducted at the grave by Harmony Lodge, No. 8, F. & A. M., with burial in Newton Cemetery.

Mr. Bulkeley was born at Galesburg, Ill., on May 24, 1875, and he was graduated from Grand Junction High School, Colorado. He received his undergraduate training at Knox College and later he did post graduate work at the University of Illinois, receiving the degree of MS in engineering.

The professional experience of Mr. Bulkeley was concerned

C. A. Bulkeley

with public service, consulting engineering, industrial engineering, research and development, air conditioning, patents, and academic work.

In 1901 he received a U. S. Treasury Department appointment as chief engineer, U. S. Court House and Post Office Building, St. Louis, Mo., having passed the required civil service examination with the highest grade. In a few months he was promoted to chief of all federal buildings in the city and later his authority extended to all federal buildings in the middle west. In 1906 he was appointed chief engineer of the Board of Education, City of St. Louis. From 1910-1917 he practiced consulting engineering with Marks & Woodwell, New York, N. Y., having supervisory charge of design and installation of all work handled by the firm. In 1917 Mr. Bulkeley became supervisory engineer, mechanical experimental division, engineering department, E. I. duPont de Nemours & Co., Wilmington, Del. During this period until 1927 he was engaged in experimental and development work of a confidential nature, largely concerned with gases, such as solvent vapors and their mixture with air and other gases, as applied to extensive processes of manufacture. From 1927-1938 Mr. Bulkeley was interested in air conditioning as a subject for special research. He joined the Niagara Blower Co., Buffalo, New York, as chief engineer, in which connection he developed a complete line of apparatus, both of unit and central plant types for all air conditioning requirements. For the past two years he was in charge of engineering and sales work in the Philadelphia office of Niagara Blower Co.

Through Mr. Bulkeley's connection with research work, much time was necessarily spent in consideration of patents. A large number of patents have been granted in the United States on inventions resulting from his work, both in air handling and other fields, and foreign patents have also been taken out in appropriate countries in all continents. Through this experience Mr. Bulkeley acquired a general engineering knowledge of patent law.

During 1936-1937 Mr. Bulkeley taught two courses in air conditioning at Columbia University open only to graduate engineers.

Mr. Bulkeley was the author of the Bulkeley Psychrometric Chart, the Bulkeley Psychrometric Chart for low temperatures, and various papers for engineering societies. He also contributed to many engineering handbooks and technical magazines.

Mr. Bulkeley was elected a Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in 1923 and at the time of his death he was a member of the Publication Committee and the Technical Advisory Committee on Psychrometry. He also served as ASHVE representative on the Blower Systems Committee of *NFPA*. He was a member of Polar Lodge, No. 79, F. & A. M., St. Louis, Mo., and Ismalia Temple, A. A. O. N. M. S. of Buffalo, N. Y.

The Officers and Council of the Society regret the loss of a distinguished member of such outstanding ability and have expressed their sympathy to Mrs. Elizabeth G. Bulkeley and a son, Merle Bulkeley, both of Long Beach, Calif., who survive.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 21 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by August 15, 1939, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ANSPACHER, T. H., Dist. Mgr., Buffalo Forge Co., Dallas, Tex. (*Advancement*)	R. K. Werner F. E. Giesecke	L. S. Gilbert C. L. Kribs, Jr.
ARKLEY, L. M., Head of Mech. Engrg. Dept., Queen's University, Kingston, Ont., Can. (*Reinstatement*)	H. G. Conn (*SAE*) C. A. Orrin (*IMME*)	D. S. Ellis (*MEIC*) S. N. Graham (*AIME*)
BARTELS, E. M., Supervisor Mech. Equip., Independent School Dist., Des Moines, Iowa	R. A. Norman Perry La Rue	F. E. Triggs C. W. Helstrom
BISHOP, J. A., Dist. Mgr., American Blower Corp., Dallas, Texas	L. S. Gilbert M. W. Brown	C. L. Kribs, Jr. E. K. Campbell, Jr.